JOINT MEETING

of

THE INSTITUTE OF MARINE ENGINEERS

and

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

Co-ordinated Alignment of Line Shaft, Propulsion Gear and Turbines*

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During the past ten years much time and effort have been devoted toward improving the design, materials, manufacturing techniques and quality control of the present advanced design marine propulsion gears. A large amount of money also has been invested in improved manufacturing equipment to obtain a higher degree of accuracy and quality of the gears. To realize the full benefits of the modern marine propulsion gears and obtain long, trouble-free operation, full consideration should be given to attaining and maintaining good internal alignment of the meshing pinions and gears. In the majority of cases when gear tooth trouble develops it occurs in the second-reduction element. This has been attributed to many causes, such as (*a*) insufficient or improper lubrication; (*b*) improper design or inaccuracies in manufacture; (*c*) inadequate gear foundation support or improper chocking; (*d*) misalignment caused by ship's hull deflexions; (*e*) excessive torsional, lateral or axial vibrations due to propeller, shafting and foundation design.

The authors now believe there is another major cause of gear tooth trouble resulting from internal misalignment in the second-reduction gear mesh which is due to an improperly designed line shaft and/or its improper alignment to the bull gear. Extensive studies, made possible by computer analysis, show how external bending moments and internal alignment of the propulsion gear are affected by such variables as the distance between the gear and first line shaft bearing, distance between line shaft supports, stiffness of shafting, wear down of supporting bearings, and deflexions of hull due to various factors.

A scientific and practical approach to the alignment of the steam turbines to avoid high speed flexible coupling and turbine vibration difficulties is also described in this paper.

FORMER LINE SHAFT GEAR ALIGNMENT SPECIFICATIONS

In the past the line shaft-gear alignment specifications of the authors' company were outlined in a simple statement as follows : With the ship water borne, the free-hanging bull gear shaft flange should be lined up to the free-hanging line shaft flange. The flanges should be in line athwartship within 0·002in.; the gear flange should be offset low by an amount equal to the calculated vertical rise of the gear journals, approximately 0·020 to 0·030in. The flanges should have a gap at the bottom from 0·000 to 0·003in. as illustrated in Fig. 1.

With this simple specification the only requirement of the gear manufacturer was to calculate and estimate the amount the centres of the two bull gear shaft journals would rise between the engine room temperature at the time of installation, and when hot during full power operation. This was accomplished by measuring the vertical expansion of various gear units while on test in the factory. From these data the vertical rise of the gear shaft journals, due to the thermal expansion, can be estimated fairly accurately.

The bull gear shaft journals also rise an additional amount, in their bearing oil clearances, when going from rest to full power operating speed. This amount is readily determined by resolving the forces on the gear, either mathematically or graphically, by making a bearing reaction diagram. When the gear shaft is offset low, in relation to the line shaft, by an mount equal to the calculated thermal expansion, plus journal rise due to bearing reaction position as shown in Fig. 1(*a*), the line shaft and gear flanges will be in line when in the hot operating condition (Fig. 1(*b*)). If the actual thermal expan-

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The Minutes of the Meeting at which this paper was presented appear on page 132 of the March 1960 TRANSACTIONS, Vol. 72.

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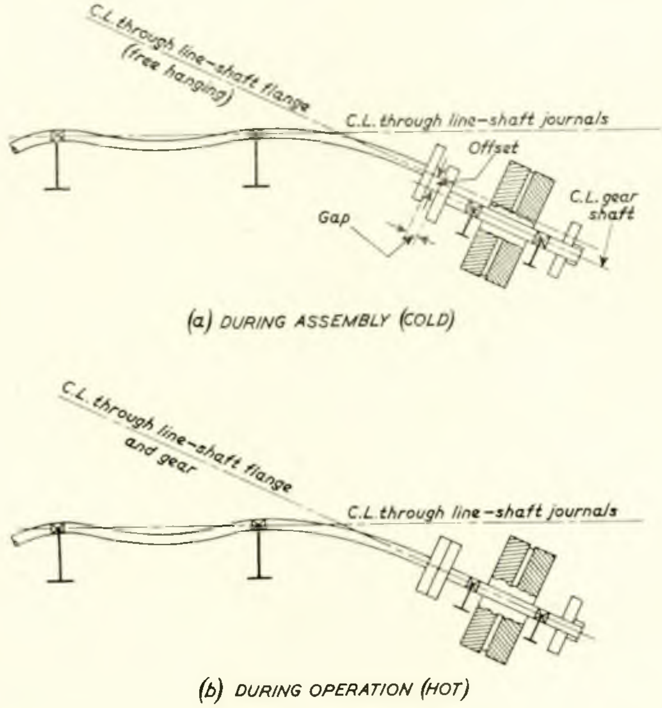


FIG. 1—*Alignment of bull gear to free hanging line shaft*

sion of the gear housing and foundation is equal to the estimated amount, then no bending moment will be present at the bolted flanges. The static bearing load reactions on the bull gear bearings will be the same as though the gear was not connected to the line shaft but just resting in its bearings.

OPERATING RESULTS WITH GEARS INSTALLED BY FORMER ALIGNMENT PRACTICE

Based on available records, experience and studies, the simple gear to line shaft alignment method previously described produced satisfactory results in some cases, providing the following conditions prevailed :

a) The main thrust bearing was located on the forward end of the bull gear, with no long thrust shaft and collar overhung on the aft end to unbalance the static forward and aft bull gear bearing load reactions.

b) The first line shaft bearing was spaced correctly from the aft bull gear bearing to give sufficient flexibility.

c) The gear was installed to a properly established target, that is, line shaft was free hanging or correctly compensated for, and not temporarily supported with proper relationship to gear disregarded.

d) The vertical rise of the bull gear journals, from the cold to the hot operating condition, was approximately the amount the gear was offset low in the cold assembly condition.

Records indicate there are many merchant ship propulsion gear units in operation where the second-reduction pinion and gear teeth have worn excessively by pitting and undercutting which originally started on the ends of the helices. In the authors' opinion this was due in many cases to inadequate alignment consideration between the gear and line shafting.

In a typical case, pitting is first noticed at one end of the tooth face, usually the aft end of the aft helix, on one second-reduction pinion. In most cases this is due to a misalignment that concentrates the load towards the ends of helices. In the early stages the pitting is more severe (large craters) at the very ends, gradually becoming less severe until it disappears about 2, 3 or 4 in. from the ends of the teeth. If the end loading and resulting pitting are observed at this time and there is an availability period, then the usual method of correcting, to arrest the pitting, is by scraping the proper bearing to compensate for the misalignment and produce a more uniform tooth contact distribution. However, in most cases the beginning of pitting at one end is not noticed or is not given proper attention. With continued operation in this condition at full power, the pitting will gradually spread across the entire face width and transfer the damage to the other pinion. The end result is undercutting, with the dedendum regions of the pinion and gear teeth being worn away, to a greater or lesser degree, into a conjugating profile and matching helix angles for the misaligned condition. The teeth then will carry the load, if not badly undercut and weakened, without further undercutting or destruction of the tooth surfaces as long as the particular misaligned condition is not changed appreciably and the lubricating system is clean. The depth of undercutting will vary from approximately 0·005in. to as much as 0·060in. or more, depending on the amount of misalignment between the pinion and the gear, length of operation, torques transmitted and other factors.

Prior to and during World War II the authors' company manufactured many C-2(6,000-s.h.p. normal) and C-3(8,500-s.h.p. normal) propulsion gears. These gears were installed in ships built at many different shipyards, using various methods of alignment between the bull gear and the line shafting. All gear units were full load torque tested in the factory at full speed and uniform tooth contact across the face width was established before shipment from the factory. However, operating records of a large number of these propulsion gear units indicated that uniformity of contact distribution across the helices on the second-reduction tooth meshes was lacking and heavy end tooth loading of each helix existed. This was, in several cases, severe enough to start pitting at the ends and required correction of internal alignment of the pinions and gear by scraping the proper second-reduction pinion bearings. It now appears that in many installations such a condition was caused by improper alignment between the bull gear and the line shafting.

PROPOSED METHOD OF LINE SHAFT AND GEAR ALIGNMENT

To reduce or eliminate the possibility of an improperly aligned line shaft, and thus prevent internal misalignment between the bull gear and mating pinions, more complete gear to line shaft alignment specifications should be issued by the gear manufacturer. The specifications should limit the difference between the vertical static downward loading on the fore and aft bull gear bearings and give calculated numerical values to various considerations that must be used when calculating the relative position of the bull gear bearings to the line shaft bearings. This information also should include gaps, sags, and offsets between all mating shaft flanges for the cold original installation. All bearing load reactions should be calculated for the assembled propeller, line shaft, and bull gear. The gear-to-line shaft alignment specification data should be available to the shipbuilder early so that these can be taken into consideration when designing the line shaft and positioning the shaft bearing pedestals. Furthermore, a weighing technique, such as calibrated jacks, should be used to check reactions accurately and quickly after all line shaft components are assembled and the ship is waterborne.

INTERNAL MISALIGNMENT OF GEARS DUE TO UNEQUAL LOADS ON BULL GEAR BEARINGS

Since the pinions and gears in a propulsion gear unit are usually supported in bearings with oil clearances, the shaft journals will be in various positions within the confines of the bearing oil clearance depending upon the direction of rotation of the shaft journal and on the magnitude of the forces acting on each journal.

When a gear unit is flexibly connected to its driving and driven members, so in the hot operating condition no external bending moments are imposed on the pinions or gears, as illustrated in Fig. 1(*b*), the normal driving forces on the teeth will position the pinions and gears in their bearings parallel to each other. Uniform tooth contact distribution will exist under these conditions, if the pinion and gear rotors are symmetrical, that is, with no large overhung weights. If, however, the design of the gear is of a type that has the main thrust collar integral with or bolted to the bull gear shaft on the aft end, the bull gear shaft, with its bolted thrust shaft, will not be symmetrical about the centre of the bull gear. In such a case, the centre of gravity of the bull gear shaft assembly is not midway between the forward and aft bearings; therefore, the static forward and aft bull gear bearing reactions will be unequal. When the load is unequal by an appreciable amount, the result will be internal misalignment of the pinions and the bull gear, and alignment of gear to line shaft must be accomplished by more complete analysis of the overall line shaft assembly.

When unequal bearing reaction forces are resolved with gear tooth driving forces, the forward and aft bull gear journals will be forced into different resultant directions in their bearing clearances. The meshing pinions and gear will then be operating in a crossed-axes condition and tooth-contact distribution will not be uniform across the face width. Instead, the tooth load will be more concentrated at one end of each helix and gradually taper off to little or no tooth contact at the opposite ends of the helices. The degree of tooth load concentration at one end of the helices will depend on the amount of resultant misalignment.

The one exception to the foregoing is when the axes of the two pinions are diametrically opposite and in the same horizontal plane with the axes of the bull gear as shown in Fig. 2. The tangential driving pressures of each pinion are

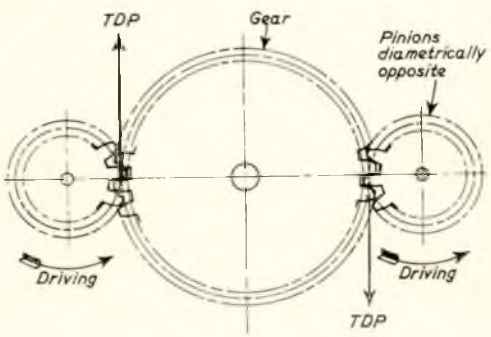


FIG. 2—*Single horizontal plane gear*

exactly opposite in direction, and when they are transmitting the same amount of torque the net resultant of forces on the bull gear, due to the pinions, is zero.

The bull gear bearing reactions, in such a case, will depend only on the weight of the bull gear plus or minus the forces resulting from connecting the gear to the line shaft. The forward and aft bull gear shaft journals will not be forced into different resultant directions, even though the static gear bearing reactions are unequal by a large amount, provided that perfect athwartship or sidewise alignment exists. In such a pinion and bull gear arrangement it is therefore only necessary that the bull gear shaft be aligned with the line shaft so that its journals are always down in their bearing clearances with a minimum static load, in the order of 1,000 to 2,000lb. on its minimum loaded bearing and a maximum of 150lb. per sq. in. unit load on the maximum loaded bearing.

BEARING REACTION LOAD DIAGRAM

To find the position of the shaft journals in their respective bearing oil clearances, under various load conditions, a so-called bearing reaction diagram is constructed.

Fig. 3 shows a typical bearing reaction diagram of the second-reduction pinions and gear, when transmitting 100 per cent torque (full power). In the diagram only the dead weight of the bull gear is used, assuming it is not bolted to the line shaft. To obtain the effect of the two pinions in the upper plane driving the bull gear, the normal driving pressure (NDP) of each pinion must be added vectorially

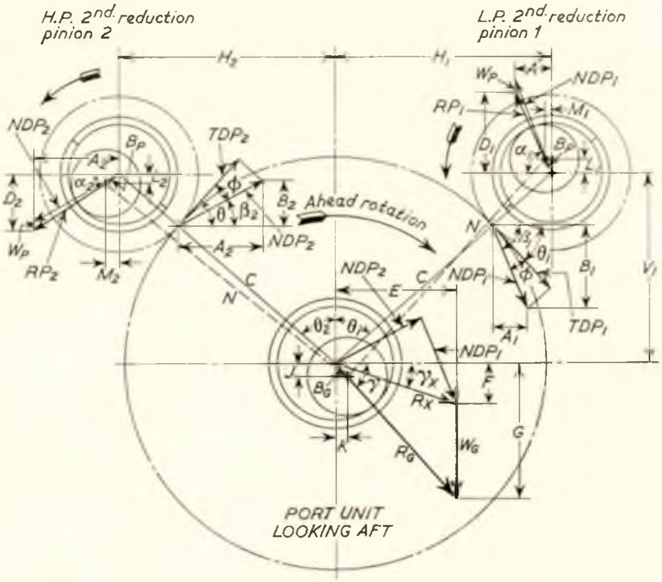


FIG. 3—*Bearing reaction diagram with equal bearing loads*

Given values

Centre distance (C), in.	79·667
Pressure angle (ϕ), deg.	20
Pitch diameter, pinion (Dp), in.	18·333
RPM (pinion), $RPMp$	923
Pinion weight (Wp), lb.	− 2,625
Gear weight (W_G), lb.	−40,000
½ diametral oil clearance, pinion (Bp), in.	0·008
½ diametral oil clearance, gear (B_G), in.	0·010

	Pinion 1	Pinion 2
HP = Horse power	5,100	4,900
H (see diagram), in.	− 60·00	+ 60·00
V (see diagram), in.	52·409	52·409

Calculated values

	Pinion 1	Pinion 2
$TDP=\dfrac{(126{,}000)(HP)}{(Dp)(RPMp)}$	36,487 lb.	37,976 lb.
$NDP=TDP/\cos\phi$	38,829 lb.	40,413 lb.
$\theta=\tan H/V$	− 48·86333 deg.	48·86333 deg.
$\beta=\theta-\phi$	− 68·86333 deg.	28·86333 deg.
$A=NDP\times\cos\beta$	14,002 lb.	35,393 lb.
$B=NDP\times\sin\beta$	− 36,216 lb.	19,508 lb.
$D=Wp-B$	33,592 lb.	− 22,133 lb.
$\alpha=\tan^{-1} D/A$	67·37293 deg.	− 32·02031 deg.
$Rp=A/\cos\alpha$	36,393 lb.	41,743 lb.
$L=Bp\times\sin\alpha$	0·0074 in.	− 0·0042 in.
$M=Bp\times\cos\alpha$	0·0031 in.	0·0068 in.

	Gear
$E=\Sigma A$	49,394 lb.
$F=\Sigma B$	− 16,708 lb.
$G=F+W_G$	− 56,708 lb.
$\gamma_X=\tan^{-1} F/E$	− 18·68856 deg.
$R_X=E/\cos\gamma$	52,143 lb.
$\gamma=\tan^{-1} G/E$	− 48·94311 deg.
$R_G=E/\cos\gamma$	75,203 lb.
$J=B_G\times\sin\gamma$	− 0·0075 in.
$K=B_G\times\cos\gamma$	0·0065 in.

This diagram shows total resultant forces on teeth and on two bearings of each rotor at 100 per cent torque, and assumes weight of each respective rotor splits 50-50 between its bearings.

(R_X) when determining the total resultant force (R_G) on the bull gear journals and bearings. The dotted lines connecting the centres of each pinion journal to the centres of the gear journals are the lines of centres between each pinion and the gear when the gear unit is operating at full load torque. It will be noted that the dotted lines are longer and not in the same plane as the full lines connecting the centres of the pinion and gear bearing bores. During manufacture of the gear housings, the axes of the bearing bores of mating elements are checked with arbors for being in a plane and for equal centre distances; therefore, they are the reference datums. When the gear unit is transmitting power and there is no appreciable external bending moment effect on the gear, the pinion and gear weights and driving forces are divided equally between the forward and aft journals and both journals will be at the same reaction point. The dotted lines of journal centres will be the same length at each end and in the same plane. Therefore, the pinion and gear axes are parallel and good uniform tooth contact is obtained.

If, however, there is an external bending moment exerted on the bull gear shaft by a coupled line shaft, as shown in Fig. 4, it can produce unequal load reactions on the forward

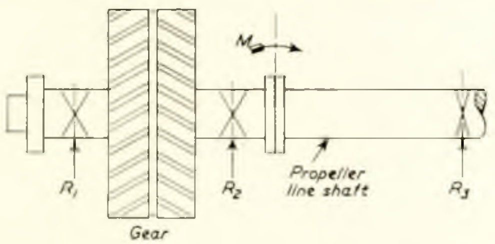


FIG. 4—*Coupled bull gear and line shaft*

and aft bull gear bearings (R_1 and R_2). In such a case it is necessary to study the total resultant bearing reactions on the forward and aft bull gear bearings separately.

Fig. 5 shows the method and calculations used in determining the effect of unequal bearing reactions on the forward and aft bull gear bearings. On both diagrams in this figure $R_x/2$ is one-half of the total force exerted on the gear journals by the two driving pinions and is in the same direction (angle γ_x) as in Fig. 3. In this case, calculations of the forces

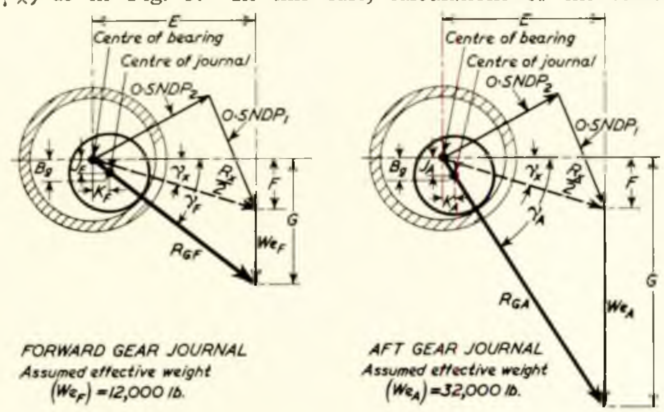


FIG. 5—*Bearing reaction diagram with unequal bearing loads*

Bg ($\frac{1}{2}$ diametral oil clearance) = 0·010

	Forward journal	After journal
We = assumed loads	12,000	32,000
E = 0·5 E (Fig. 3)	24,697	24,697
F = 0·5 F (Fig. 3)	−8,354	−8,354
G = F + We	−20,354	40,354
γ_x (Fig. 3), deg.	−18·68856	18·68856
$\frac{R_x}{2} = F/\cos\gamma_x$, lb.	26,072	26,072
$\gamma = \tan^{-1} G/E$, deg.	−39·49326	−58·53283
$R_G = E/\cos\gamma$	32,003	47,311
$J = B_G \sin\gamma$	−0·0064	−0·0085
$K = B_G \cos\gamma$	0·0077	0·0052

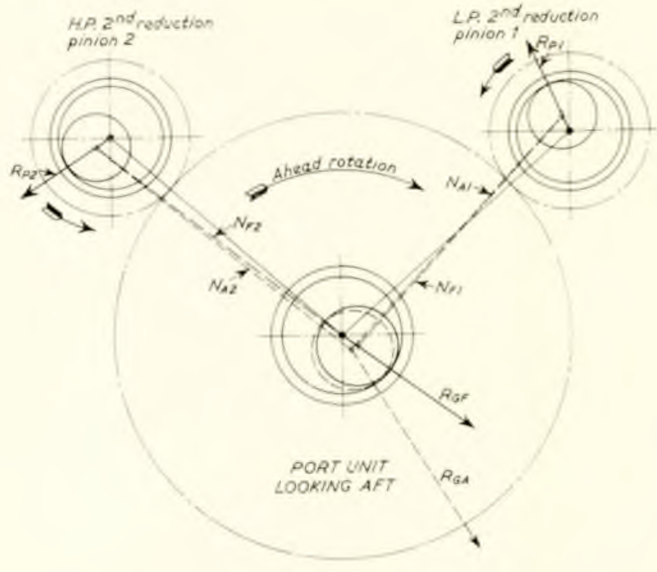


FIG. 6—*Diagram showing relation of cocked gear and meshing pinions*

exerted by the connected line shaft, with a given alignment condition, produced a reaction or effective downward weight, We_A on the aft bearing of 32,000lb. and on the forward bearing 12,000lb., We_F. When the effective downward weights are added vectorially to their respective bearings, as shown in Fig. 5, the resultant forces, R_{GA} and R_{GF}, are neither equal nor in the same direction. This indicates that the bull gear will be skewed in its bearing clearances. The amount of skewing or crossed-axis condition can be determined by using one-half the diametral oil clearance, *Bg*, and calculating the co-ordinates, *J* and *K*, of the forward and aft journals, with respect to the centres of the bearing bores as a datum.

Fig. 6 shows second reduction pinions and their relation to the skewed bull gear. The co-ordinates of the centres of the pinion journals, *L* and *M*, with respect to centres of the bearing bores can be calculated in a similar manner as previously explained. The co-ordinates will be the same for both the forward and aft pinion journals as there is no bending moment imposed on the pinions being flexibly connected to the first reduction gears. They will be in the same position as shown in Fig. 3. The forward gear journal is shown by solid lines and the aft journal in dotted lines in their respective positions as determined and shown in Fig. 5.

It will be noted that there are two contributing factors causing misalignment in this case. The centre distances *Nf*

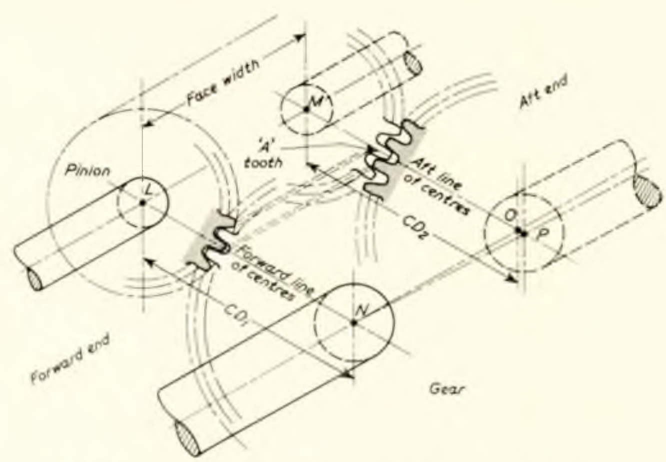


FIG. 7—*Diagram showing effect of unequal centres*

and *Na* on the forward and aft ends of the pinion and gear are unequal and the centres of the two pinions and two gear journals are not in the same plane. This produces a crossed-axes condition and one end of the mating tooth face is open. The effect is similar to a helix angle mismatch. In this case the aft centre distance is greater than the forward. The effect is to increase the backlash on the aft end, thus producing an opening in the helix angle mismatch on this end equal to one-half of the effective increase in backlash.

Fig. 7 illustrates the effect of unequal centre distances by means of a simplified sketch showing a pinion and gear having perfectly machined spur teeth with the line of centres in a horizontal plane. They are cut by two perpendicular planes, one through the forward journals and the other through the aft journals. In the sketch the centre distance CD_1 has a value such that, at tight mesh, no backlash exists at the forward end. Assuming that the pinion and gear were supported with the centres of their journals at *L, M, N* and *O*, then CD_1 would equal CD_2 and the teeth would be at tight mesh at both ends and across the face width. When journal centre *O* is moved to *P* in the same plane, it increases the centre distance at the aft end by an amount *OP*. It can be seen with tooth *A*, located on the line of centres, that when the centre distance CD_2 is increased, that is, the gear teeth are moved away from the pinion by an amount *OP*, an opening or backlash between meshing teeth is produced. The opening is equal on each side of the pinion tooth *A* and is one-half of the total increase in backlash. It is apparent that the opening will be a maximum on one end and uniformly tapering off to zero at the other end. The effect is quite similar for single or double helical gear teeth. The opening will be zero on the end of each helix and increase uniformly at each crossover on each helix, for a given misalignment. The helix angle must of course be considered in the calculations for the backlash and opening.

Fig. 8 illustrates the effects of the centres of the four

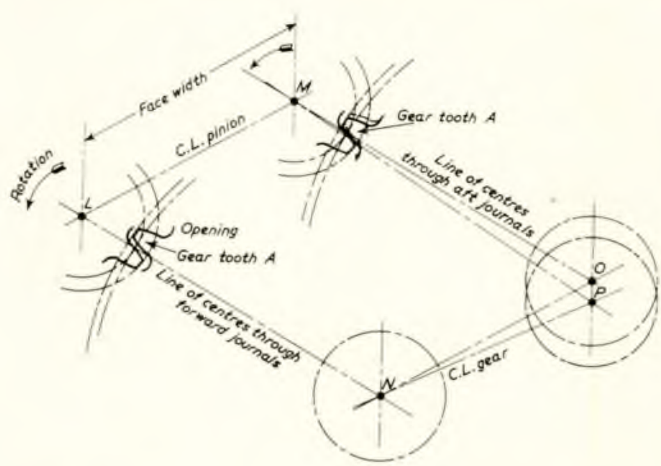


FIG. 8—*Meshing diagram showing effect of out of plane of journals*

pinion and gear journals not being in a common plane. The centre distances between the forward and aft journals are now the same, that is, *LN, MO* and *MP* are equal, and there is the usual amount of uniform backlash along the face width. It should be noted that if the centres of the four journals are in a plane *LMNO*, the gear teeth will be parallel to the pinion teeth and uniform tooth contact will exist across the face width. That is, gear tooth *A* will be in the position shown by the dotted lines at the aft end. When the centre *O* is lowered any given amount to position *P*, the gear tooth *A* at the aft end, shown by solid lines, will contact a pinion tooth at the aft end but not at the forward end. In this case the maximum opening is at the forward end and gradually tapers off at a uniform rate to zero at the aft end. The pinion and gear are said to be " out of plane " by an amount *OP*.

Since it is easier to study and illustrate this effect when the centres of the three journals, *L, M,* and *N* are, as in this case, in a level or horizontal plane, the distance *OP* is referred to also as the amount "out of level." Although it may not be shown too clearly in Fig. 8, it can be noted however that gear tooth *A* at the aft end in the plane *MOP* is lowered exactly the same amount as distance *OP*. This happens when the aft end gear journal centre *O* is lowered out of plane to position *P;* the entire aft end of the complete gear rotor is then lowered and is pivoting about point *N* or line *LN*. The effect is similar on either single-helix or double-helical gears. The opening or mismatch on each helix becomes increasingly greater on each helix with increasing misalignment.

In this example the mismatch or opening caused by the greater centre distance at the aft end of Fig. 7 is compensated by the out of level condition of Fig. 8. In most cases of misalignment, this compensating effect is present; however, in some cases the two effects are additive. It is therefore necessary when calculating, to keep the plus and minus signs in proper order. It will be noted also that the effect of unequal centre distance does not influence the mismatch between the meshing pinion and gear teeth nearly as much as the effect of the same amount of out of plane condition. For example, if centre distances were unequal by 0·002in., it would cause an opening of 0·0007in. under certain specific conditions. Under the same conditions an out of level of 0·002in. will produce an opening of 0·002in.

Fig. 9 shows the geometric relationship of the pinion and bull gear shaft centres and Table I the calculations required to obtain the amount of opening between the teeth at one end of the helix due to a given unequal load (20,000lb.) on the fore and aft bull gear bearings. The out of level and unequal centre distances, ΔL and ΔC respectively in Table I, show the effect on pinions 1 and 2. It will be noted that for pinion 1 the two effects compensate for one another whereas on pinion 2 they are additive. The resulting openings, or mismatch, normal to the tooth surfaces for each pinion, show that for the same amount of unequal load on the bull gear bearings, with the greater load on the aft bearing, pinion 1 is open 0·0003in. at the aft end, and is referred to as the "least affected" pinion, and pinion 2 is open 0·0010in. on the forward end of each helix, and is referred to as the " worst affected " pinion.

CONSIDERATIONS AND REQUIREMENTS OF PROPOSED METHOD OF ALIGNMENT

If all the external forces acting upon the propulsion gear are given due consideration and the gear unit is designed properly, accurately manufactured and installed, no gear operating difficulty should be experienced for many years. The authors realize that external forces from various sources do exist and have to be considered. Analysis and studies indicated that the forces resulting from improper line shaft alignment to the bull gear shaft, plus incorrectly designed line shaft and arrangement of shaft supports, can have a great influence upon the satisfactory operation of the propulsion gear. The possibility of imposing excessive external bending moments and subsequent unequal bearing loads on the bull gear shaft journals is greatest when the line shaft is relatively short, stiff, and supported only in the stern tube and strut bearings, such as on destroyers. In cases such as this, the relatively large permissible wear-down of stern tube bearings before replacement can cause changes in the vertical height relationship of line shaft journals so the bearing reactions are changed significantly. The gear bearing reactions are also affected adversely due to the relatively short distance from stern tube to gear bearings.

On large powered bulk carriers, with engine rooms far aft, that have relatively short, stiff line shafts, the possibility of exerting excessive external bending moments which affect the internal alignment of the gear is also great.

As mentioned previously, one of the main requirements of a satisfactory gear line shaft alignment is one that produces nearly equal static load reactions in direction and amount on

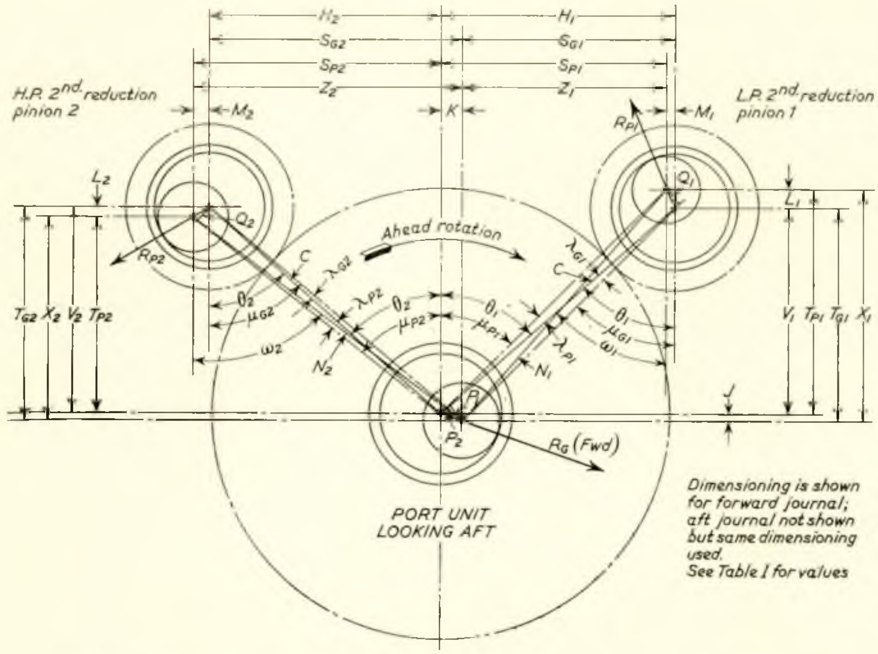


FIG. 9—*Geometric relationship of pinion and bull gear shaft centres with unequal load on bull gear bearings*

TABLE I.—FORMULAS AND NUMERICAL VALUES FOR BEARING REACTION DIAGRAM (FIG. 9)
(Based on values given and calculated in Figs. 3 and 5)

	Pinion 1		Pinion 2	
$S_P = H + M$, in.	−59·99692		60·00678	
$T_P = V + L$, in.	52·41638		52·40476	
$\mu_P = \tan^{-1} \frac{S_P}{T_P}$, deg.	−48·85787		48·86883	
$\lambda_P = \mu_P - \theta$, deg.	0·00546		0·00551	
$Q = C \tan \lambda_P$ (pinion out-of-level), in.	0·00758		0·00766	
	Forward journal	Aft journal	Forward journal	Aft journal
$S_G = H + K$, in.	−59·99228	−59·99478	60·00772	60·00522
$T_G = V - J$, in.	52·41536	52·41753	52·41536	52·41753
$\mu_G = \tan^{-1} \frac{S_G}{T_G}$, deg.	−48·85623	−48·85624	48·86353	48·86118
$\lambda_G = \mu_G - \theta$, deg.	0·00710	0·00709	0·00206	− 0·00215
$P = C \tan \lambda_G$ (gear out-of-level), in.	0·00987	0·00986	0·00029	− 0·00299
Total out-of-level $= P + Q$, in.	0·01745	0·01744	0·00795	0·00467
$Z = H + K + M$, in.	−59·98921	−59·99170	60·01450	60·01200
$X = V - J + L$, in.	52·42274	52·42491	52·41112	52·41329
$\omega = \tan^{-1} \frac{Z}{X}$, deg.	−48·85078	−48·85078	48·86904	48·86668
$N = \frac{Z}{\sin \omega}$ (actual centre distance), in.	79·66712	79·67042	79·67851	79·67806
$^*\triangle L = TL_F - TL_A$, in.	0·00001		0·00328	
$^*\triangle C = N_F - N_A$, in.	−0·00330		0·00045	
†Effective backlash (EB), in.	−0·00120		0·00016	
*Mismatch $(M_J) = \triangle L + EB$, in.	−0·00119		0·00343	
*Opening normal to teeth M_N, in. $\left(M_N = M_J \frac{F_W}{J_G} \cos \phi\right)$	−0·00034		0·00098	

NOTES: * A plus answer means forward end open.
A minus answer means aft end open.
† "Effective backlash" is the opening between two meshing teeth due to an increase in centre distance between the two parts. It is actually one half of the backlash resulting from this increased centre distance. It is calculated as follows:

$$EB = C \triangle \text{inv } \phi$$

$$\triangle \text{ inv } \phi = \text{inv } \phi_1 - \text{inv } \phi$$

$$\phi = \cos^{-1}\left[\cos \phi - \cos \phi \left(\frac{\triangle C}{C + \triangle C}\right)\right]$$

the forward and aft bull gear bearings when the gear is in its full load operating position. However, there are many other limits that must not be exceeded, and a complete list would include the following:

1) For the majority of gears studied, the difference in static fore-and-aft bull gear bearing reactions at full power operating conditions should not exceed in the order of 5,000 to 15,000lb. in the vertical component. In the horizontal direction it should not be more than 1,000 to 2,000lb. (in hot operating condition only).

2) All line shaft and bull gear bearings should have a downward load in the order of 1,000 to 2,000lb. minimum under all conditions of operation.

3) Unit load on all bearings should be within good practice limits. That is, ring or wick-oiled line shaft bearing pressures should not exceed approximately 40lb. per sq. in. Pressure lubricated gear bearings should not exceed 150lb. per sq. in. when in the static condition.

4) Total combined stress in all line shaft components should not exceed safe values. It is felt that for materials in general use, a bending stress of approximately 15,00lb. per sq. in. should not be exceeded.

5) All of the foregoing requirements must be met when the ship is waterborne, with the machinery in either the cold static or hot operating conditions, and with the original settings and clearances of the stern tube bushings and line shaft bearings, as well as with the maximum allowable wear of these parts.

With reference to item 1, the authors realize that it is practically impossible to obtain and maintain an ideal equal static load condition on bull gear bearings. They therefore calculate and specify a maximum difference in loading that will limit the mismatch or opening between teeth of meshing pinions and gear to approximately 0·0002in. per ft. of face width per helix on the ends of each helix. This is calculated for each main propulsion gear unit and is given to the shipbuilder as part of the gear manufacturer's instructions and recommendations for aligning the bull gear to the line shaft. The calculations required to obtain this limit of unequal loading, although not complex, are quite lengthy. If programmed for the IBM-704 computer or equivalent, results can be quickly and accurately obtained whenever required.

To illustrate more clearly the calculation and use of this maximum static bearing load, difference on bull gears, and also other factors, a few actual line shaft arrangements will be reviewed :

a) A large tanker having eight total bearings and a bolted-on main thrust aft of the bull gear.

b) A bulk cargo ship having eight total bearings and a main thrust on the forward end of the gear.

c) A midship engine room installation, having twelve total bearings and main thrust integral on the aft end of the gear.

d) A twin-screw navy ship (port shaft) with seven bearings and main thrust integral on the aft end of the gear.

SHAFT ALIGNMENT RECOMMENDATIONS ON A LARGE TANKER

On a recent, high power, single-screw bulk carrier, the following gear and line shaft alignment installation data were furnished by the authors' company to the shipyard for guidance in aligning the gear to the line shaft. (The main thrust bearing, mounted in a separate housing with a short, bolted thrust shaft, was furnished by the gear manufacturer and installed immediately aft of the gear.)

1) With all parts, such as gear foundation, gear housings, and line shaft bearing pedestals at operating temperature and bull gear shaft journals in their full load operating positions, the difference between the vertical static downward loadings on the forward and aft bull gear bearings, due to weight of bull gear and external bending moments imposed by the line shaft, should not exceed approximately 14,000lb.

2) The 14,000-lb. difference in loading can be on either the forward or aft bull gear bearing.

3) The line shaft should be aligned to the bull gear in the athwartship direction, so that no significant (less than 5,000lb.) forces are imposed on the bull gear bearings in the horizontal plane.

Considerations

4) Due to bearing reaction position at rated condition, the centre of the bull gear shaft will rise vertically approximately 0·004in. and move athwartship approximately 0·010in. with a bull gear diametrical bearing clearance of 0·028in.

5) It is estimated that because of thermal expansion, the centre line of the bull gear shaft will rise vertically approximately 0·020—0·025in. (using the ship's inner bottom as a datum) when going from 80 deg. F. temperature to the operating temperature. This figure and any differential expansion of the line shaft bearing supports must be considered.

6) Bending stress in the bull gear shaft must not exceed 15,000lb. per sq. in. when bolted to the line shaft.

7) Bearing load reaction on either bull gear bearing in static cold or hot condition should not exceed 101,400lb. (based on 150lb. per sq. in. unit load on projected area).

8) With the thrust shaft bolted to the bull gear shaft, the free hanging flange of thrust shaft will sag 0·014in. below the centre line of the bull gear journals. There will be a gap of 0·006in. between the bottom of the thrust shaft flange and a line perpendicular to the centre line of the journals, as shown in Fig. 10.

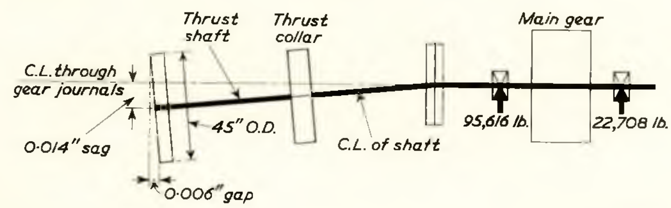


FIG. 10—*Gap and sag of bull gear and thrust shaft assembly*

The 14,000-lb. maximum difference in static loading between the forward and aft gear bearings, given in item 1, was determined from curves in Fig. 11. Based on 0·0002in. per ft. of face width per helix, an opening of 0·0003in. was considered allowable in this case, and note that it is limited by the low pressure pinion, 2, the worst affected.

The curves were established from calculated data shown in Table II. This table shows the mismatch, or opening, at the ends of the helices of each pinion and gear for three assumed values of unequal load (10,000, 20,000 and 30,000lb.) between the forward and aft bull gear bearings.

Tables II and III give the output data from the computer with which it is possible to plot the curves in Fig. 11 and construct a pinion and gear bearing reaction diagram to show diagrammatically the position of each journal in its respective bearing.

The same equations are used as shown for the example in Table I.

Fig. 12 is the bearing reaction diagram for this unit based on the foregoing data for 100 per cent torque. In the calculations, total weights of the bull gear and pinions are used and the total driving pressures were assumed acting on one bearing of each element. This can be done if it is assumed that all loads are equally divided between the forward and aft journals. When this is done it is necessary to divide the final resultant by 2 to obtain a value for the forward and aft journals, respectively. For example, the gear reaction (total resultant force) is 211,387lb., half of which is 105,694lb. or the load on each bull gear bearing. Since only the total weight of the gear was used, any external forces acting on the bull gear bearings will increase or decrease these values depending on how the line shaft is designed and aligned to the gear shaft.

Fig. 13 is a similar diagram, wherein it was assumed that the bull gear had 30,000lb. more static downward loading on its aft bearing than on the forward bearing, as a result of certain alignment to the line shaft, wherein there is a static load of 70,500lb. on the aft bearing and 40,500lb. on the

TABLE II.—UNEQUAL LOAD ON BULL GEAR BEARINGS

	Weight	*G*	Gamma	*RG*	*J*	*K*
Forward	−50,500·0	−78,908·2	−47·01470	107,867·6	−0·01024	0·00933
Forward	−45,500·0	−73,908·2	−45·14104	104,265·7	−0·00992	0·00988
Forward	−40,500·0	−68,908·2	−43·13561	100,783·2	−0·00957	0·01022
Aft	−60,500·0	−88,908·2	−50·40230	115,384·5	−0·01079	0·00892
Aft	−65,500·0	−93,908·2	−51·93330	119,279·8	−0·01102	0·00863
Aft	−70,500·0	−98,908·2	−53·36658	123,254·8	−0·01123	0·00835

Mismatch due to unequal loading

	Second-reduction h.p. pinion 1	Second-reduction l.p. pinion 2
Unequal load = −10,000·0		
Fwd LVL	0·024050	0·013242
Aft LVL	0·023814	0·012423
Fwd CTR	103·36626	103·37611
Aft CTR	103·36706	103·37624
Dif LVL	0·000236	0·000819
Dif CTR	−0·000793	−0·000132
Eff BL	−0·000288	−0·000046
Opening	−0·000015	0·000218
Unequal load = −20,000·0		
Fwd LVL	0·024160	0·013690
Aft LVL	0·023692	0·012053
Fwd CTR	103·36582	103·37602
Aft CTR	103·36741	103·37628
Dif LVL	0·000468	0·001637
Dif CTR	−0·001590	−0·000266
Eff BL	−0·000581	−0·000096
Opening	−0·000032	0·000434
Unequal load = −30,000·0		
Fwd LVL	0·024263	0·014166
Aft LVL	0·023569	0·011704
Fwd CTR	103·36534	103·37590
Aft CTR	103·36773	103·37632
Dif LVL	0·000694	0·002462
Dif CTR	−0·002392	−0·000419
Eff BL	−0·000873	−0·000150
Opening	−0·000050	0·000652

TABLE III.—DATA FOR SECOND-REDUCTION MESH BEARING REACTION DIAGRAM

Pinion weight = −7,800·0 Gear weight = −95,000·0
Pinion r.p.m. = 674·0 Centre distance = 103·35980

	Pinion 1	Pinion 2
H	−55·0000	55·0000
V	87·5110	87·5110
HP	12,875·0	12,125·0
TDP	90,090·0	84,842·1
NDP	95,896·1	90,309·9
Theta	−32·14905	32·14905
Beta	−52·18887	12·10923
A	58,790·1	88,300·5
B	−75,761·3	18,944·9
D	67,961·3	−26,744·9
Alpha	49·13848	−16·85075
RP	89,861·1	92,261·9
L	0·00832	−0·00319
M	0·00720	0·01053
Q	0·01052	0·01061

	Gear
E	147,090·6
F	−56,816·5
G	−151,816·5
Gamma	−45·90581
RG	211,385·6
J	−0·01005
K	0·00974

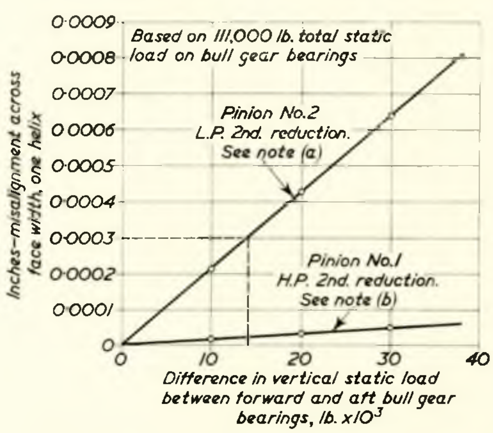


FIG. 11—*Curves of tooth face mismatch* versus *unequal bearing loads*

Note. Heaviest load can be on forward or aft gear bearings:

a) Forward end of helix is open when heavier load is on aft bearing; aft end of helix is open when heavier load is on forward bearing.
b) Aft end of helix is open when heavier load is on aft bearing; forward end of helix is open when heavier load is on forward bearing.

forward bearing. Note that the aft journal is in a different position in its bearing than the forward journal; thus, the gear is operating in a cross-axes (skewed) position with respect to the pinions. The mismatch is worse on the low pressure, second-reduction pinion (pinion 2) than on the high pressure

Given		
Gear weight (W_G), lb.		−95,000
Pinion weight (Wp), lb.		− 7,800
Pressure angle (ϕ)		20° 2′ 23·36″
Pitch diameter, pinion (Dp), in.		26·7166
Centre distance (C), in.		103·3598
$RPMp$ (pinions)		674
½ diametral oil clearance, pinion (Bp), in.		0·011
½ diametral oil clearance, gear (B_G), in.		0·014
	Pinion 1 (h.p.)	Pinion 2 (l.p.)
Horse power (HP)	12,875	12,125
H, in.	− 55	+ 55
V, in.	87·511	87·511
Calculated		
$\theta=\tan^{-1}\frac{H}{V}$, deg.	− 32·14095	32·14905
$TDP=\frac{126{,}000\times HP}{Dp\times RPM}$, lb.	90,090	84,842
$NDP=TDP\div\cos\phi$, lb.	95,896	90,310
$\beta=\theta-\phi$, deg.	− 52·18887	12·10923
$A=NDP\times\cos\beta$, lb.	58,790	88,301
$B=NDP\times\sin\beta$, lb.	− 75,761	18,945
$D=W_p-B$, lb.	67,961	− 26,745
$\alpha=\tan^{-1}(D\div A)$, deg.	49·13848	− 16·85075
$RP=A\div\cos\alpha$, lb.	89,861	92,262
$L=B_p\times\sin\alpha$, in.	0·00832	− 0·00319
$M=B_p\times\cos\alpha$, in.	0·00720	0·01053
	Gear	
$E=\Sigma A$, lb.	147,091	
$F=\Sigma B$, lb.	− 56,816	
$G=F+W_G$, lb.	− 151,816	
$\gamma=\tan^{-1}(G\div E)$, deg.	− 45·90581	
$RG=E\div\cos\gamma$, lb.	211,387	
$J=B_G\times\sin\gamma$, in.	− 0·01005	
$K=B_G\times\cos\gamma$, in.	0·00974	

NOTE: Above values calculated on IBM-704, using Programme MGE-403.

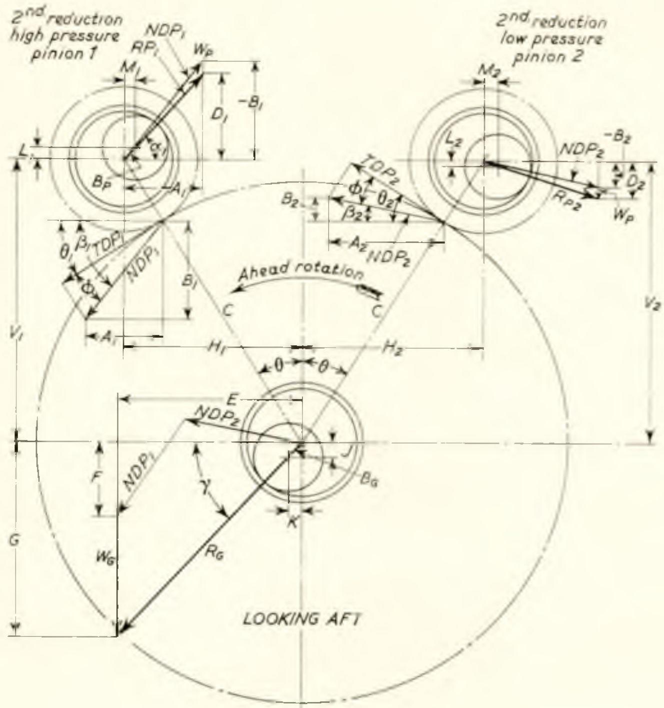


FIG. 12—*Bearing reaction diagram with equal bearing loads*

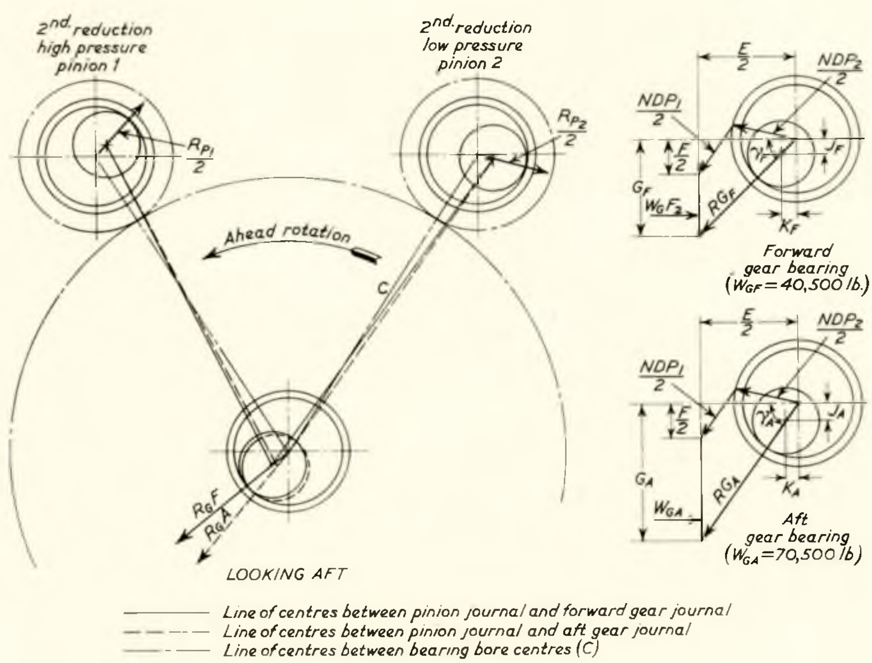


FIG. 13—*Bearing reaction with 30,000-lb. unequal bearing load at 100 per cent torque*

	Forward bearing	Aft bearing
G, lb.	68,908	98,908
γ, deg.	43·1356	53·3666
R_G, lb.	100,783	123,255
J, in.	0·00957	0·01123
K, in.	0·01022	0·00835

Based on output from IBM-704 Programme MGE-403 pinion 1, as shown by the curves in Fig. 11, where the mismatch or opening at 30,000lb. difference in gear bearing loading is 0·00065in. on the low pressure pinion and only 0·00005in. on the high pressure pinion.

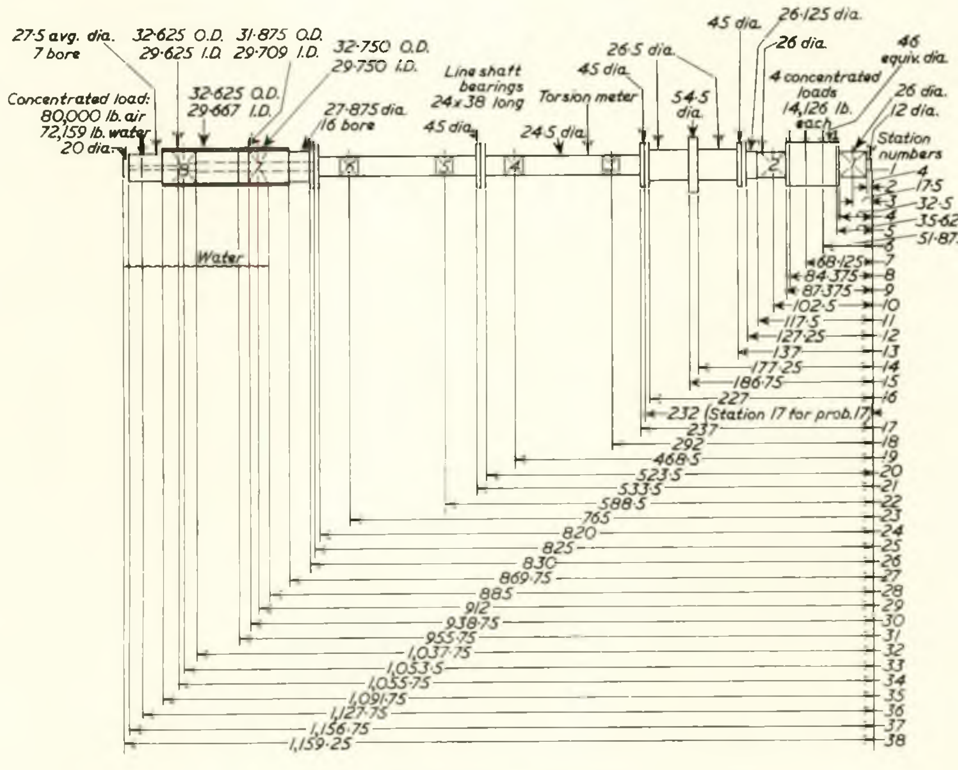
The misalignment of the high pressure, second-reduction pinion (pinion 1), due to its geometric relationship as shown in Fig. 13, is only slightly affected by large differences in loading between the forward and aft bull gear bearings. It too, however, is operating in a crossed-axes condition, as shown by the figures near the bottom of Table II. With a difference in load of 30,000lb. on the bull gear bearings it will be noted that (pinion 1) second-reduction high pressure is operating with one journal 0·000694in. out of plane (Dif. LVL) with respect to the other three journals. The centre distance is 0·002392in. greater at the aft end than at the forward end. However, since the out-of-plane condition practically compensates for the difference in centre distance, the net result is that a satisfactory tooth contact exists and the misalignment opening is only 0·00005in., shown at bottom of Table II.

As previously stated, in item 4 under "Considerations", the bull gear journals will rise 0·004in. vertically, from the static position, when the gear is transmitting full power. Item 5 states that bull gear journals will also rise approximately 0·020 to 0·025in. from the cold to the hot position due to the thermal expansion of the lower gear housing and gear foundation. Therefore the total rise at operating condition will be approximately 0·024 to 0·029in. This value must be considered when aligning the gear and line shaft in the cold condition in order to obtain optimum internal gear and pinion alignment and bearing loading in the hot operating condition.

Item 8 in the list of "Conditions" and Fig. 10 show the results of gap and sag calculations made on the bull gear and main thrust shaft assembly. It will be noted that with the thrust and bull gear shafts bolted, and the gear supported in its journals, the free hanging thrust shaft flange will sag 0·014in. below the centre line through the bull gear journals; it will have a 0·0006in. gap on a 45-in. outside diameter flange, as shown; and the aft bull gear bearing reaction will be 95,616lb. and the forward bearing reaction 22,708lb.

In order to obtain the line shaft characteristics and study various possible means of obtaining approximately equal loads on the bull gear bearings when in the hot operating condition, calculations were made on the complete line shaft arrangement. A unique computer programme originated by the Boston Naval Shipyard was used in making the calculations. Fig. 14 shows a sketch of the line shaft arrangement. All dimensions originate from a common point, the forward end of the bull gear shaft, with station numbers assigned at each bearing at each concentrated load and at each significant change in diameter of shaft.

It is assumed that all bearing reactions act at a point on the fore and aft centre line of each bearing, with the exception of the aft stern tube. At the stern tube it is assumed that the bearing load is concentrated at one shaft diameter forward of the aft end of the tube. Actually, the bearing loads are distributed on areas of the bearing. Assuming they act at point supports does not significantly change the calculated results except that in cases of long stern tube bearings the effective error may be greater, since the actual load concentration point may be some distance on either side of the assumed point, depending on the angle at which the stern tube was actually bored.



All dimensions are in inches

	lb.
Total weight gear assembly	= 95,000
Shaft section as shown	= 38,497
Concentrated load gear	= 56,503
Divided by 4	= 14,126 each load

FIG. 14—*Propeller line thrust and bull gear shaft arrangement*

Bearing Reaction Load Variation Resulting from Change in Setting of Bearings

Table IV lists the bearing load reactions when all bearings are in a straight line and the influence on the load reaction on each bearing per 0·001in. offset in the vertical direction. The values listed were taken from the computer output data and tabulated in a more convenient form. Below the table an example is given showing how the influence numbers are used to calculate the bearing reactions for any position of bearing setting.

With all bearings on a straight line in the cold non-operating condition, the bearing reaction loads will be as shown in Table IV. During the operating or hot condition, when numbers 1 and 2 bull gear bearings will rise vertically 0·030in., the resulting bearing reactions will be shown by the following calculations :

Number 1 bearing = 37,715 + (30) (1,551) − (30) (2,390) = 12,545lb.

Number 2 bearing = 73,169 − (30) (2,390) + (30) (3 827) = 116,279lb.

Number 3 bearing = 30,068 + (30) (1,042) − (30) (1,953) = 2,738lb.

Load reactions on the other bearings may be calculated in a like manner.

The calculations in Table IV are based on the assumption that there is no clearance in the bearings. This means that when a bearing unloads it will exert a negative load on the shaft; that is, the bearing reaction is downward rather than upward. The idea that a bearing exerts a negative load is useful as a transistory step in the calculations to arrive at a final decision. When a negative load is shown it indicates that the bearing may actually be unloaded at that point or is loaded in the upper half, depending on the amount of oil clearance in the bearing.

It will be noted that bearing number 7, forward stern tube bearing, has a reaction of −40,384lb. This indicates that such a downward force is necessary to bring the centre of this journal in a straight line with all other journals. In a good line shaft arrangement, no minus bearings reaction should exist, as it indicates that the bearing will be unloaded, and can produce a long span of unsupported shaft which may cause shaft whipping. To overcome such a condition it would be necessary to lower bearing number 8 the proper amount below the centre of bearing number 7. It also may be necessary to lower bearing number 7 a certain amount below bearing number 6, in order to have a downward loading on all bearings; that is, an upward or plus reaction. For this example, they would neglect the negative reaction on bearing number 7, since there are four babbitted bearings between the bull gear and the stern tube bearings. Although the influence of the stern tube bearing alignment on the loading of gear bearings numbers 1 and 2 will be relatively insignificant, it is important that the stern tubes be bored at correct heights and angles, so a proper load and as good a distribution as possible will be obtained.

Table V shows bearing reactions under various conditions of original setting of the line shaft and calculated by use of influence numbers in Table IV. Column 1 shows the bearing load reactions on the bearings when all bearings are in line in an assumed cold condition. Then, in the hot operating condition, when bearings numbers 1 and 2 rise 0·030in., the bearing load reactions will be as shown in column 2. It will be noted that the static downward loading on bearing number 2, aft bull gear bearing, increased from 73,169lb. in the cold condition to 116,279lb. in the hot condition. This obviously will produce a bad misalignment between the second-reduction pinions and the gear, particularly on number 2 pinion when the gear unit is in operation. However, the misalignment will not be present while in the static condition, because, regardless of the amount of unequal load on the forward and aft bull gear bearings, if both bull gear shaft journals are resting in their bearings, the bull gear will not be skewed or misaligned with the meshing pinions. The skewed position of the bull gear in its bearing clearance exists only in the operating con-

Table IV.—Bearing reactions and influence numbers

Bearing number	1 Forward gear	2 Aft gear	3 1st line shaft	4 2nd line shaft	5 3rd line shaft	6 4th line shaft	7 Forward stern tube	8 Aft stern tube
				Reaction loads with all bearings in straight line				
	37,715	73,169	30,068	20,526	18,966	36,925	−40,384	151,339
				Bearing reactions influence numbers (lb. per 0·001 in. bearing rise)				
1	+1,551	−2,390	+1,042	−289	+99	−18	+6	−1
2	−2,390	+3,827	−1,953	+736	−252	+46	−16	+4
3	+1,042	−1,953	+1,724	−1,403	+680	−123	+44	−10
4	−289	+736	−1,403	+2,466	−2,047	+747	−268	+59
5	+99	−252	+680	−2,047	+2,515	−1,606	+784	−173
6	−18	+46	−123	+747	−1,606	+2,486	−2,391	+860
7	+6	−16	+44	−268	+784	−2,391	+3,232	−1,391
8	−1	+4	−10	+59	−173	+860	−1,391	+652

Example: When bearing number 1 is raised 0·001in., its reaction is increased 1,551lb. The reaction on number 2 bearing is decreased 2,390lb. It is increased 1,042lb. on number 3, decreased 289lb. on number 4, increased 99lb. on number 5, and so on, as shown the table.

Table V.—Bearing reactions under various settings of line shaft bearings in the cold condition

Column	1	2	3	4	5	6
	All bearings in line	No. 1 up 0·030 in. No. 2 up 0·030 in.	No. 3 up 0·0346 in.	No. 1 up 0·030 in. No. 2 up 0·030 in. No. 3 up 0·0346 in.	No. 3 up 0·040 in. No. 4 up 0·016 in.	No. 1 up 0·030 in. No. 2 up 0·030 in. No. 3 up 0·040 in. No. 4 up 0·016 in.
Bearing number	Cold	Hot	Cold	Hot	Cold	Hot
1	37,715	12,545	73,768	48,598	74,771	49,601
2	73,169	116,279	5,595	48,705	6,825	49,935
3	30,068	2,738	89,718	62,388	76,580	49,250
4	20,526	33,936	− 28,018	− 14,608	3,862	17,272
5	18,966	14,376	42,494	37,904	13,414	8,824
6	36,925	37,765	32,763	33,509	43,957	44,797
7	− 40,384	− 40,684	− 38,861	− 39,162	− 42,912	− 43,212
8	151,339	151,429	150,993	151,083	151,883	151,973

dition when the second-reduction pinions are driving the bull gear in their regular manner. In this condition the normal driving pressure on the pinions, coupled with the unequal downward loadings on the bull gear bearing, produces the skewed effect of the bull gear, as described previously.

Since the line shaft and gear alignment condition as shown in columns 1 and 2, Table V, is not satisfactory, it was determined using the influence numbers that when bearing number 3 was raised 0·0346in. the reaction loads on numbers 1 and 2 gear bearings would be equal under the hot operating condition. This is shown in columns 3 and 4, Table V. It will be noted that under these conditions number 4 bearing will be unloaded as indicated by the minus bearing reaction sign. To eliminate this condition and obtain a more satisfactory hot operating bearing reaction loading on bearings numbers 1, 2, 3, 4, 5 and 6, it was necessary to raise number 3 bearing 0·040in., and number 4 bearing 0·016in. in the cold condition. Columns 5 and 6, Table V, show load reactions on all bearings in the cold and hot conditions. This meets all the specified requirements for good line shaft and gear alignment, with the exception that number 3 bearing may be overloaded, as designed, and the negative load reaction at number 7 bearing might cause a stern tube problem.

Gap and Sag Method of Alignment of Gear and Line Shaft

Fig. 15 shows two sketches of line and bull gear shafting, illustrating two methods of alignment using the "gap" and "sag" measurements. The significance of the terms "gap" and "sag" as used here is illustrated in the two-shaft arrangement sketches in this figure; "sag" denoting vertical distance between centres of flanges, and "gap" the amount the bottoms of mating flanges are more open than the tops of the flanges. (The terms gap and sag are also used in Fig. 10 to denote the amount the free-hanging thrust shaft flange sags below the centre line through the gear journals and how much gap there is between the bottom of the outside diameter of the flange and a line perpendicular to the centre line of journals.) Fig. 16 shows curves of calculated gap and sag relationship between the thrust shaft flange and the line shaft flange. Curves *A* show the gap and sag relationship for a cold setting which will produce equal bearing reactions on the numbers 1 and 2 bull gear bearings when the centres of the bull gear shaft journals rise vertically 0·030in. in the hot operating condition. The cold setting is based on the thrust shaft being temporarily supported so the centre of its flange *T* is in line with the bull gear shaft centres, as shown in Fig. 15 (*A*).

Curves *B*, Fig. 16, give the gap and sag relationship for a cold setting which will produce equal bearing reactions on the numbers 1 and 2 bull gear bearings when the centres of the bull gear shaft journals rise vertically 0·030in. in the hot operating condition. In this case, the cold setting is based on the thrust shaft hanging unsupported and allowed to sag, as shown in Fig. 15 (*B*). The only difference between sketches *A* and *B* is that the thrust flange is unsupported in sketch *B* and allowed to sag its natural amount, which is shown by Fig. 10.

When the bull gear, with its bolted thrust shaft, is set in

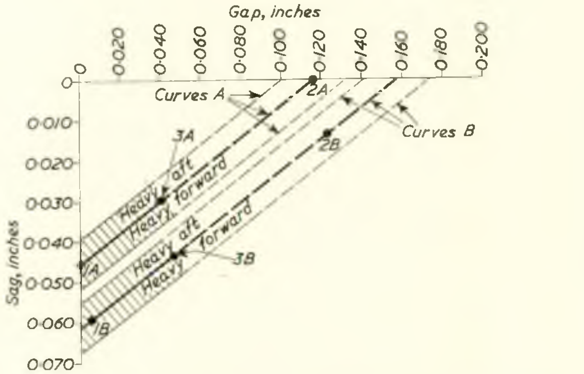


FIG. 16—*Curves of calculated gaps and sags for equal bearing loads*

Assumptions

1) All line shaft journal centres are on a line.
2) Line shaft is supported such that the centre of its free-hanging flange is on line with its journals, and the flange face is square with this line.
3) Gear journals will rise vertically upward 0·030in. more than line shaft journals when going from a cold (room temperature) to a hot, full load, operating condition.

Notes

a) Curves *A* give the gap and offset relationship for a cold setting, so that forward and aft bearing reactions will be equal when centre of line shaft gear journals move up 0·030in. to a hot operating condition. The cold setting is based on thrust shaft supported so the centre of its flange is in line with the line shaft gear journal centres (Fig. 15(*A*)).
b) Curves *B* give the gap and offset relationship for a cold setting, so that forward and aft bearing reactions will be equal when centres of line shaft gear journals move up 0·030in. to a hot operating condition. The cold setting is based on the thrust shaft hanging freely, unsupported, in its natural sag, as in Fig. 15(*B*) (and shown in more detail in Fig. 10).
c) If the gear is set cold to any gap and offset combination that is read in the cross-hatched area, line shaft gear bearing reactions will be equal within 14,000lb. in the hot, operating condition, and all line shaft bearings will have a downward reaction. Settings in the dotted line area will satisfy gear bearing requirements, but will result in one or more line shaft bearings being unloaded.

the cold condition to any gap and sag combination that is indicated in the cross-hatched area of curves *A* and *B*, the two bull gear bearing reactions will be within the 14,000-lb. differential limit required in the hot operating condition, and all line shaft bearings will have a positive or downward reaction. All gap and sag settings shown in the dotted line area of curves *A* and *B* will satisfy the bull gear bearing requirements but will result in one or more line shaft bearings being unloaded.

Table VI shows bearing reaction values for the various bearings with the line shaft completely assembled to the bull

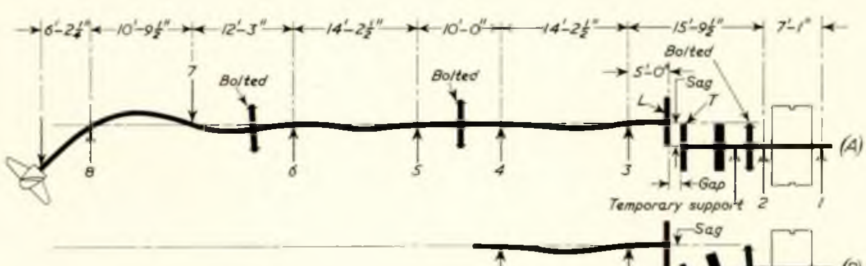


FIG. 15—*Gap and sag method of shaft alignment*

TABLE VI.—BEARING LOAD REACTIONS FOR VARIOUS GAP AND SAG CONDITIONS

Column	0·0456-in. sag: zero gap 1	2	Zero sag: 0·117-in. gap 3	4	0·030-in. sag: 0·041-in. gap 5	6
Bearing number	Nos. 1 and 2 down 0·0456 in. Cold	Nos. 1 and 2 down 0·0156 in. Hot	No. 1 up 0·558 in. No. 2 up 0·337 in. Cold	No. 1 up 0·588 in. No. 2 up 0·367 in. Hot	No. 1 up 0·165 in. No. 2 up 0·088 in. Cold	No. 1 up 0·195 in. No. 2 up 0·118 in. Hot
1	75,973	50,803	97,743	72,573	83,310	58,140
2	7,642	50,752	29,248	72,358	15,595	58,705
3	71,610	44,280	— 46,657	— 73,987	— 30,134	2,804
4	143	13,553	107,296	120,706	37,609	51,019
5	25,943	21,353	— 10,716	— 15,306	13,125	8,535
6	35,648	36,488	42,383	43,223	38,003	38,843
7	— 39,928	— 40,228	— 42,428	— 42,728	— 40,802	— 41,102
8	151,202	151,292	152,129	152,219	151,526	151,616

gear shaft and the gear and thrust shaft set to various gap and sag combinations as determined by curves *A* and *B*, Fig. 15. Columns 1 and 2 show the bearing reactions that would exist in the cold and hot conditions respectively, if the bull gear and thrust shaft had been set to zero gap and 0·046in. low as shown at point *1A* in Fig. 16 with the end of the thrust shaft supported. The same conditions would exist with a free-hanging thrust shaft set to a gap of 0·006in. and sag of 0·060 in., as indicated by point *1B* in Fig. 16.

Columns 3 and 4, Table VI, show the bearing reactions that would exist in the hot and cold conditions if the gear had been set to zero sag and 0·117-in. gap as shown by point *2A*, Fig. 16, with the end of the thrust shaft supported; or with a sag of 0·014in. and gap of 0·123in. as indicated by point *2B* with free-hanging thrust shaft. Since these alignment settings produce large negative bearing reaction values on numbers 3 and 5 line shaft bearings, it is considered unsatisfactory, and is illustrated here as a matter of information only. Actually, with clearance in the bearings, the loadings would not be as shown.

Columns 5 and 6 in Table VI show the bearing reactions for an intermediate setting on the gap and sag curves, as indicated by points 3A and 3B, Fig. 16. It will be noted that in this case, all bearings from numbers 1 through 6 have a downward loading in the hot condition.

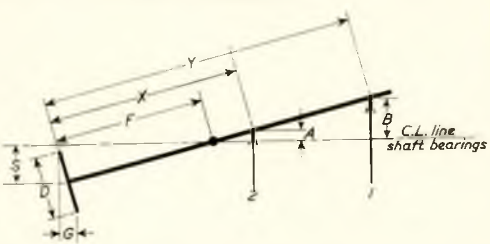


FIG. 17—*Geometric relation method for calculating gear journal positions*

X = distance from aft end of flange to aft gear bearing (2)
Y = distance from aft end of flange to forward gear bearing (1)
D = diameter of flange
S = sag (centre of flange below centres of line shaft bearings)
G = gap (opening at bottom of flange)
A = amount number 2 bearing is raised
B = amount number 1 bearing is raised
F = distance from aft end of flange to fulcrum of system

$$F = \frac{SD}{G}$$

$$A = (X - F)\left(\frac{G}{D}\right) = \frac{XG}{D} - S$$

$$B = (Y - F)\left(\frac{G}{D}\right) = \frac{YG}{D} - S$$

When centre of flange is on centre line of line shaft bearings, sag = 0:

$$\frac{A}{X} = \frac{B}{Y} = \frac{G}{D} \qquad A = \frac{X}{Y}B \qquad G = \frac{D}{Y}S$$

Fig. 17 shows the geometric relationships used in calculating bull gear bearing journal movements off the straight line for a given gap and sag reading.

Although alignment of shafting made by using the free-hanging thrust shaft method, as shown in Fig. 15(*B*), and using curves *B* in Fig. 16, eliminates chocking and rechocking the temporary support position on the thrust shaft each time the bull gear position is changed (in order to maintain it and the bull gear shaft in a straight line), it places a considerable load on number 2 bearing, which may cause excessive foundation deflexions, and hence an error in alignment when load is relaxed later on in the bolted-up condition.

Effect of Horizontal Static Forces on Internal Alignment of Gear

Up to this point only the effects of vertical static forces were considered. It is of course obvious that horizontal static forces also will affect the bull gear bearing reactions and hence the internal alignment of the bull gear with its meshing pinions. With a line shaft bearing displaced horizontally in

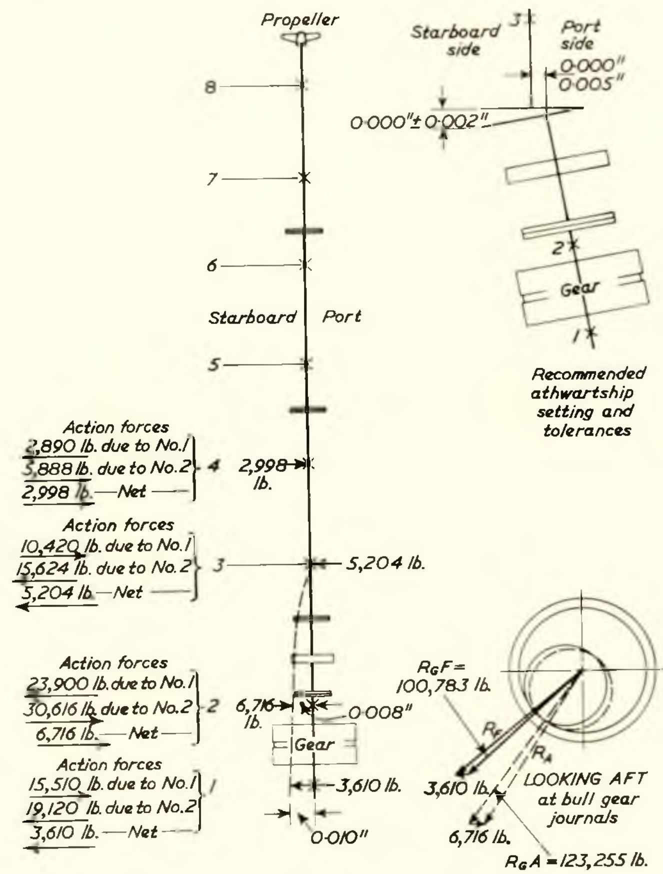


FIG. 18—*Plan view sketch of line shaft arrangement*

the order of 0·005 to 0·010in., the resultant forces are usually relatively insignificant, and will have little effect on the final resultant bearing reactions. The magnitudes of the forces do however depend on the characteristics of the line shaft and the bearing influence numbers involved. The resultant forces on the bearings, due to moving journals in a horizontal direction, may be determined by using the bearing influence numbers in the same manner as used to calculate the forces in the vertical direction. This is understandable when it is realized that the bearing influence numbers are in reality the spring constants of the line shaft, as referred to each bearing journal support. The difference is that when used in the horizontal plane, the weight of the shaft is not involved.

Referring to Fig. 13, it will be seen that at full power operation the forward gear journal moves to starboard 0·010in., as indicated by dimension K_F, while the aft journal moves to starboard only 0·008in., dimension K_A.

Fig. 18 is a plan view of the line shaft, thrust and bull gear shaft with all journal centres assumed in line, as shown by the solid line. The dotted line shows the athwartship positions of the centres of the forward and aft gear journals. These are displaced to the starboard 0·010 and 0·008in. respectively, due to resultant positions of the journals in their bearing clearance under full power operation, with 30,000lb. assumed unequal static vertical load between the two bull gear bearings.

Using the influence numbers in Table IV, the net force on each of the first four bearings, due to the athwartship movement, was calculated. These are action forces, not reaction, therefore their direction is opposite from that indicated by the influence numbers, which refer to reactions. For bearing number 1, moving it 0·010in. to starboard produces a force on its journal of 15,510lb. to port. Moving number 2 bearing 0·008in. to starboard produces a force of 19,120lb. to starboard on bearing number 1. This results in a net force of 3,610lb. acting to starboard on bearing number 1. The

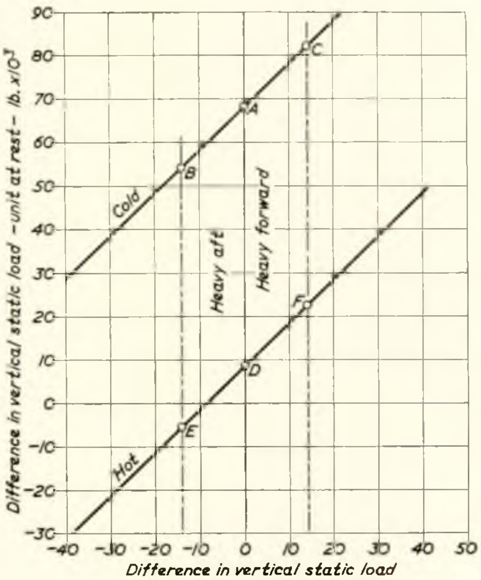


FIG. 19—*Bearing reaction load check weight curves*

Notes

1) Plus values on "difference in load" scales indicate heavier load is on forward bearing.
2) Minus values indicate that heavier load is on aft bearing.
3) These curves are based on the assumption that the gear unit rises 0·026in. due to thermal expansion, and gear journals rise 0·004in. when operating at full power.

net forces on bearings numbers 2, 3 and 4 were calculated in a similar manner to show they too are affected slightly by the athwartship movement of bearings numbers 1 and 2.

The sketch in the lower right hand corner of Fig. 18 shows the bull gear bearing and position of forward and aft gear journals under 30,000-lb. unequal static vertical load condition, as shown in Fig. 13. When the horizontal forces are added, the effect is to increase slightly the amount of skewing of the bull gear. As can be seen from the sketch in Fig. 18, the resultant forces due to unequal static vertical loading on the forward and aft journals are 100,783 and 123,255lb. respectively. When the horizontal vectors, of 3,610 and 6,716lb. respectively, are added to the much greater resultant forces caused by vertical loading, they have practically no effect on the final position of the journal centres.

The recommended athwartship setting of the bull gear to line shaft is shown in the sketch in the upper right hand corner of Fig. 18. This should limit the static forces on the bull gear bearings in the horizontal plane to 5,000lb. maximum for this case.

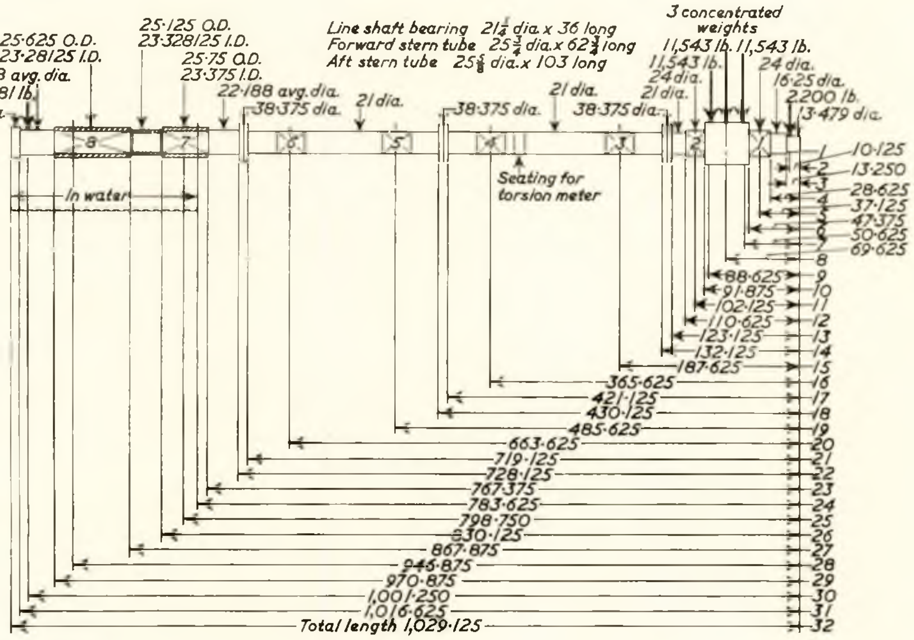
Regardless of the method used in aligning the bull gear to the line shaft, an attempt should be made, if at all possible, to measure the actual bearing reactions on the bull gear bearings and the first two line shaft bearings, after the assembly and alignment are complete. This should be done with the plant at room temperature and also hot, that is, just after a dock trial run, for example, when the temperature of the housings is stabilized, or by circulating hot oil. The authors appreciate the problems involved in measuring the bearing reactions accurately, owing to the friction in the bearings, the flexibility of ship's beam and pedestal supports, and not being able to measure at the centre of each bearing, to mention a few. However, by proper approach and analysis it can be accomplished.

If, for example, all journals were set in line in the cold condition, the calculated bearing reactions would be as shown in column 1, Table V. Upon measuring the reaction, if it was found to be approximately 37,000lb. on number 1 forward gear bearing, and 30,000lb. on number 3 bearing, then, without measuring the aft bull gear reaction, it could be assumed that it is approximately 73,000lb. It would, of course, be more accurate to measure the reaction on both bull gear bearings and know their actual values. When measuring these reactions, the curves in Fig. 19 can be used as a guide to determine if reactions will be equal in the hot operating condition. In the two curves shown, the upper labelled "cold" is used when weighing bearing reactions in the cold condition and assumes gear bearings will rise 0·030in. when in the hot, full power operating condition. For instance, in order to have equal reactions on the forward and aft bull gear bearings in the hot operating condition, 0 on X-axis, the forward bearing should weigh 68,272lb. heavier than the aft bearing in the cold condition, point A on the curve. With the allowable 14,000lb. unequal load taken into consideration, the difference between fore and aft cold may be from 54,272lb. (point B on curve) to 82,272lb. indicated at point C.

When the bearing reactions are weighed under hot conditions, the lower curve labelled "Hot" is used. It will be noted that the forward bull gear bearing reaction should be 9,104lb. heavier (point D) than the aft, in order that the two reactions will be equal when both gear journals have risen the additional 0·004in. at full load operating condition. Allowing for the permissible 14,000-lb. difference in loading, calculations indicate that the aft gear bearing may be 4,896lb. heavier (point E) than the forward bearing; or the forward bearing 23,104lb. heavier (point F) than the aft bearing. Under these conditions the final reactions will be within the specified limits.

SHAFTING AND GEAR ALIGNMENT ANALYSIS ON BULK CARGO SHIP WITH MAIN THRUST BEARING ON FORWARD END OF GEAR

Fig. 20 shows a sketch of the line shaft arrangement for a bulk cargo ship, where the main thrust bearing is on the



All dimensions are in inches

Propeller weight in air = 56,800 lb.

$$\text{Propeller weight in water} = 56{,}800 \times \frac{490 - 64}{490} = 56{,}800 \times 0{\cdot}86939 = 49{,}381 \text{ lb.}$$

	Lb.
Total weight of low speed gear and shaft assembly	= 61,700
Low speed gear and shaft only	− 59,500
Concentrated weight for thrust collar	= 2,200

Calculated weight for complete line shaft, as shown = 211,064 lb. (including concentrated weights).

Completely in air = 222,752 lb. (including concentrated weights).

	Lb.
Weight of low speed gear and shaft	= 59,500
Shaft (as sectioned above—calculated)	= 24,872
Concentrated weight for low speed gear	= 34,628
Concentrated weight for each web (1/3)	= 11,543

FIG. 20—*Arrangement view of bull gear and line shaft*

forward end of the bull gear, and is dimensioned the same as described for Fig. 14.

The following data were furnished by the authors' company to the shipbuilder for his guidance on this installation in aligning the gear to the line shaft to avoid internal misalignment of the bull gear with the meshing second-reduction pinions :

1) When all parts, such as gear foundations, gear housing and line shaft pedestals, are at operating temperature and the second-reduction gear journals are in their full load operating position, the difference in the vertical static downward load, due to the weight of the bull gear and external bending movement imposed by the line shaft, should not exceed 5,000 lb. between the forward and aft bull gear journals.

2) The larger load can be on either the forward or aft gear bearing.

3) The alignment in the athwartship direction should be such that no significant (less than 1,000lb.) forces are imposed on the bull gear bearings in the horizontal plane.

4) Calculations of the bearing reaction position, at normal rated condition, indicated that the centre of the bull gear shaft will rise vertically approximately 0·003in. and move athwartship approximately 0·008in. with a diametral bearing oil clearance of 0·026in.

5) Assuming a temperature rise of approximately 40 to 50 deg. F. of the gear foundation, oil sump and lower gear housing between cold and stable operating temperatures, it is estimated that the centre of the bull gear shaft will rise vertically approximately 0·015—0·020in. This figure and any differential expansion of the line shaft bearing supports must be considered.

6) The bending stresses in the bull gear shaft resulting from bolting to the line shaft must not exceed 15,000lb. per sq. in.

7) The bearing reaction loading on either bull gear bearing in static cold or hot condition should not exceed 54,000lb. based on a limit of 150lb. per sq. in. unit load on the projected area of the bearing.

8) The bull gear shaft flange has negligible sag, less than 0·001in., with the gear supported in its journals and flange not bolted to the line shaft.

With all bearings in line, in the as-cold condition, the resulting shaft deflexions and bearing loads as calculated are shown in Fig. 21(a). They are also shown in Table VII, first column on the left. In the hot operating condition when the bull gear will rise vertically approximately 0·020in. in relation to the line shaft bearings, the loads on the various bearings will be shown in Table VII, second column from the left.

It shows that the number 1 forward bull gear bearing and the number 3 bearing will be unloaded in the bottom half,

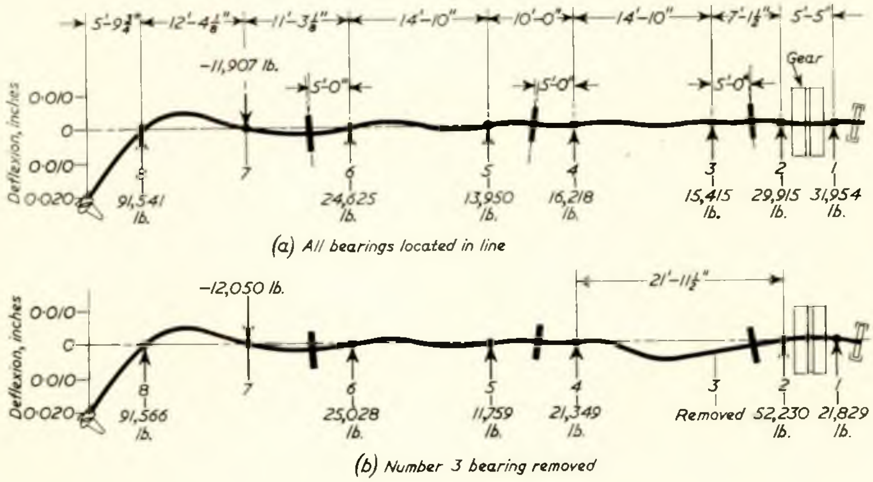


FIG. 21—*Elevation view of shafting showing deflexions and bearing loads*

TABLE VII.—BEARING REACTIONS WITH VARIOUS SETTINGS OF LINE SHAFT BEARINGS

Bearing number	All bearings in line	No. 1 up 0·020 in. No. 2 up 0·020 in.	No. 3 up 0·0175 in.	No. 1 up 0·020 in. No. 2 up 0·020 in. No. 3 up 0·0175 in.	No. 3 up 0·0186 in. No. 4 up 0·0069 in.	No. 1 up 0·020 in. No. 2 up 0·020 in. No. 3 up 0·0186 in. No. 4 up 0·0069 in.
	Cold	Hot	Cold	Hot	Cold	Hot
1	31,954	−15,006	76,614	29,664	77,406	30,442
2	29,915	128,375	−68,523	29,937	−67,776	30,666
3	15,415	−46,045	83,420	21,960	78,773	17,313
4	16,218	30,478	−6,410	7,850	1,966	16,234
5	13,950	9,010	23,610	18,670	16,406	11,474
6	24,625	25,525	22,840	23,740	25,578	26,486
7	−11,907	−12,227	−11,277	−11,597	−12,245	−12,567
8	91,541	91,601	91,429	91,485	91,602	91,658

Assumptions

(*a*) All bearings are on a straight line except as indicated.

(*b*) Bull gear bearings will rise vertically 0·020 in. going from a cold (engine room temperature) to a hot operating condition. The bearing reactions in Table VII were calculated using straight line reactions and influence numbers as shown in Table VIII.

TABLE VIII.—BEARING REACTIONS AND INFLUENCE NUMBERS

Bearing number	1 Forward gear	2 Aft gear	3 1st line shaft	4 2nd line shaft	5 3rd line shaft	6 4th line shaft	7 Forward stern tube	8 Aft stern tube
	Reaction loads with all bearings in straight line							
	31,953·8	29,914·7	15,415·4	16,218·0	13,950·3	24,624·7	−11,907·3	91,541·0
	Bearing reaction influence numbers (lb. per 0·001 in. bearing rise)							
1	2,664·2	−5,012·4	2,552·3	−291·9	100·9	−18·5	6·6	−1·2
2	−5,012·4	9,934·5	−5,625·3	1,005·3	−347·5	63·9	−22·7	4·0
3	2,552·3	−5,625·3	3,885·9	−1,293·2	552·3	−101·6	36·0	−6·4
4	−291·9	1,005·3	−1,293·2	1,419·8	−1,131·9	412·0	−146·1	26·1
5	100·9	−347·5	552·3	−1,131·9	1,358·0	−875·9	418·7	−74·7
6	−18·6	63·9	−101·6	412·0	−875·9	1,359·4	−1,218·5	379·2
7	6·6	−22·7	36·0	−146·1	418·7	−1,218·5	1,503·7	−577·8
8	−1·2	4·0	−6·4	26·1	−74·7	379·2	−577·8	250·8

as they have negative loads. The aft bearing, number 2, under this condition is excessively loaded, and the bull gear and shaft are tilted. This is unsatisfactory and indicates a condition which may exist under operating conditions if the gear and line shafting are aligned as indicated. Bearing number 7, forward stern tube bearing, is also unloaded, but is far enough removed from gear bearings so its influence will not be considered in this paper.

Columns 3 and 4 show another unsatisfactory condition where, by raising numbers 3 bearing 0·0175in. the bull gear bearings are equally loaded at the hot operating condition, but in the cold condition the aft gear bearing is unloaded in the lower half. The number 4 bearing is also unloaded.

The two right hand columns in Table VII show a satisfactory hot condition, but unsatisfactory when the gear unit is cold. Under this condition the aft bull gear bearing is unloaded by a large amount. The bearing influence numbers pertaining to Table VII are shown in Table VIII as calculated by the IBM-704 computer.

It was therefore concluded that, with the arrangement of shafting and bearings as shown in Fig. 20, it would be impossible to meet all the requirements as previously specified. Another analysis was then made with number 3 bearing removed. The shaft deflexions and bearing loads for this arrangement, with all bearings in a line, are illustrated in Fig. 21(b). The bearing load reactions for various conditions of setting of line shaft bearings are shown in Table IX. The bearing influence numbers are shown in Table X, and by

TABLE IX.—BEARING REACTIONS FOR VARIOUS SETTINGS OF LINE SHAFT BEARINGS WITH NUMBER 3 BEARING OMITTED

Bearing number	All bearings in line	No. 1 up 0·020 in. No. 2 up 0·020 in.	No. 4 up 0·0326 in.	No. 1 up 0·020 in. No. 2 up 0·020 in. No. 4 up 0·0326 in.	No. 4 up 0·0509 in. No. 5 up 0·0349 in.	No. 1 up 0·020 in. No. 2 up 0·020 in. No. 4 up 0·0509 in. No. 5 up 0·0349 in.
	Cold	Hot	Cold	Hot	Cold	Hot
1	21,829	15,233	40,003	33,407	41,086	34,490
2	52,230	61,706	23,969	33,445	23,875	33,351
3	—	—	—	—	—	—
4	21,349	15,161	53,603	47,415	38,604	32,416
5	11,759	15,563	−19,149	−15,345	8,178	11,982
6	25,028	24,328	37,351	36,651	14,219	13,519
7	−12,050	−11,802	−16,418	−16,170	−4,422	−4,174
8	91,566	91,520	92,345	92,299	90,205	90,159

Assumptions

(a) All bearings are on a straight line except as indicated.

(b) Bull gear bearings will rise vertically 0·020in. going from a cold (engine room temperature) to a hot operating condition.

NOTE: The foregoing bearing reactions were calculated using straight line reactions and influence numbers shown in Table X.

TABLE X.—BEARING REACTIONS AND INFLUENCE NUMBERS

Bearing number	1 Forward gear	2 Aft gear	3 1st line shaft	4 2nd line shaft	5 3rd line shaft	6 4th line shaft	7 Forward stern tube	8 Aft stern tube
	Reaction with all bearings in straight line							
	21,826·6	52,230·3	—	21,348·7	11,758·7	25,028·4	−12,050·2	91,566·2
	Bearing reaction influence numbers (lb. per 0·001-in. bearing rise)							
1	987·7	−1,317·5	—	557·5	−261·9	48·2	−17·1	3·0
2	−1,317·5	1,791·3	—	−866·9	452·1	−83·2	29·5	−5·3
3	—	—	—	—	—	—	—	—
4	557·5	−866·9	—	989·4	−948·1	378·2	−134·1	23·9
5	−261·9	452·1	—	−948·1	1,279·5	−861·4	413·6	−73·8
6	48·2	−83·2	—	378·2	−861·5	1,356·7	−1,217·5	379·0
7	−17·1	29·5	—	−134·1	413·6	−1,217·5	1,503·4	−577·7
8	3·0	−5·3	—	23·9	−73·8	379·0	−577·7	250·7

comparison, note that the high influence numbers for bearings numbers 1, 2, 3 and 4 in Table VIII have been reduced considerably by removing bearing number 3.

Table IX, columns 1 and 2, with all bearings in line cold, shows an unsatisfactory condition because the difference in loading between numbers 1 and 2 bearings in the hot condition is excessive. Table XI shows that with an unequal load of 5,000lb., the misalignment (opening) between the gear and number 2 pinion is 0·000217in. This is considered as being the maximum allowable misalignment for this gear; therefore, the difference between the static bull gear bearing loadings should be held within approximately 5,000lb., in the hot operating condition.

Table IX, columns 3 and 4, shows another setting, with number 4 up 0·0326in. to produce equal gear bearing reactions in the hot condition, which is also unsatisfactory owing to the large negative loading of bearing number 5 under both cold and hot conditions. A satisfactory setting of line shaft bearings is shown in columns 5 and 6 in the cold and hot conditions respectively. To obtain this satisfactory alignment

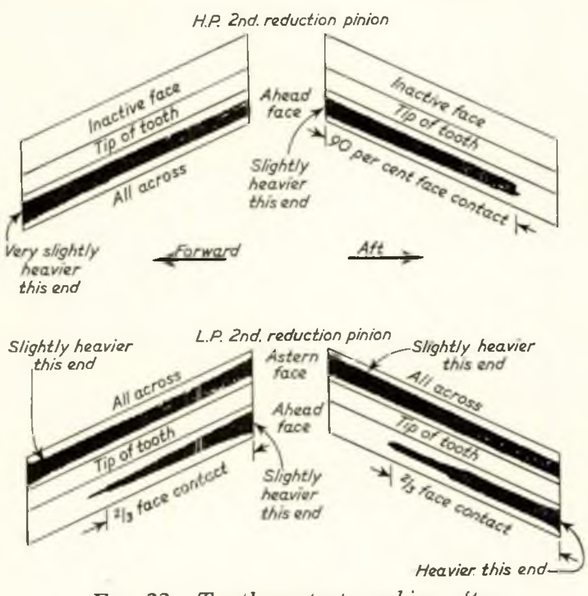


FIG. 22—*Tooth contact marking after maiden voyage*

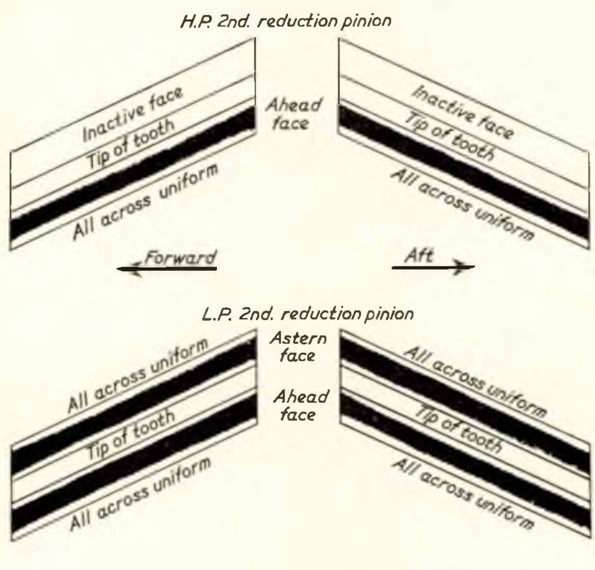


FIG. 23—*Tooth contact marking after line shaft bearing adjustment*

TABLE XI.—UNEQUAL LOAD ON GEAR BEARINGS

	Weight	*G*	Gamma	*RG*	*J*	*K*
Forward	−30,000·0	−41,901·2	−49·94912	54,738·9	−0·00995	0·00837
Forward	−25,000·0	−36,901·2	−46·33314	51,013·1	−0·00940	0·00898
Forward	−20,000·0	−31,901·2	−42·16711	47,521·9	−0·00873	0·00964
Aft	−35,000·0	−46,901·2	−53·09350	58,654·6	−0·01040	0·00781
Aft	−40,000·0	−51,901·2	−55·83723	62,724·6	−0·01076	0·00730
Aft	−45,000·0	−56,901·2	−58·24184	66,920·8	−0·01105	0·00684

Mismatch due to unequal loading

	Second-reduction h.p. pinion 1	Second-reduction l.p. pinion 2
Unequal load = −5,000·0		
Fwd LVL	0·020063	0·006254
Aft LVL	0·019971	0·005544
Fwd CTR	91·83651	91·84797
Aft CTR	91·83722	91·84790
Dif LVL	0·000092	0·000710
Dif CTR	−0·000708	0·000072
Eff BL	−0·000254	0·000023
Opening	−0·000048	0·000217
Unequal load = −15,000·0		
Fwd LVL	0·020120	0·007073
Aft LVL	0·019861	0·004928
Fwd CTR	91·83569	91·84800
Aft CTR	91·83783	91·84780
Dif LVL	0·000259	0·002145
Dif CTR	−0·002139	0·000198
Eff BL	−0·000778	0·000069
Opening	−0·000154	0·000656
Unequal load = −25,000·0		
Fwd LVL	0·020121	0·008018
Aft LVL	0·019739	0·004893
Fwd CTR	91·83475	91·84797
Aft CTR	91·83836	91·84769
Dif LVL	0·000382	0·003624
Dif CTR	−0·003615	0·000278
Eff BL	−0·001315	0·000100
Opening	−0·000276	0·001104

condition it is necessary to raise number 4 bearing 0·0509in., and number 5 bearing 0·0349in. vertically above all other bearings in the cold condition. In this particular case the complete shaft calculations and alignment data were not available during the installation of the shafting and the propulsion gear. Therefore, the shipyard aligned all line shaft and bull gear bearings in line, according to their usual practice, and then lowered number 3 bearing 0·010in., so it was carrying no load. This alignment would be the same as that shown in column 1, Table IX, which was previously stated as being unsatisfactory.

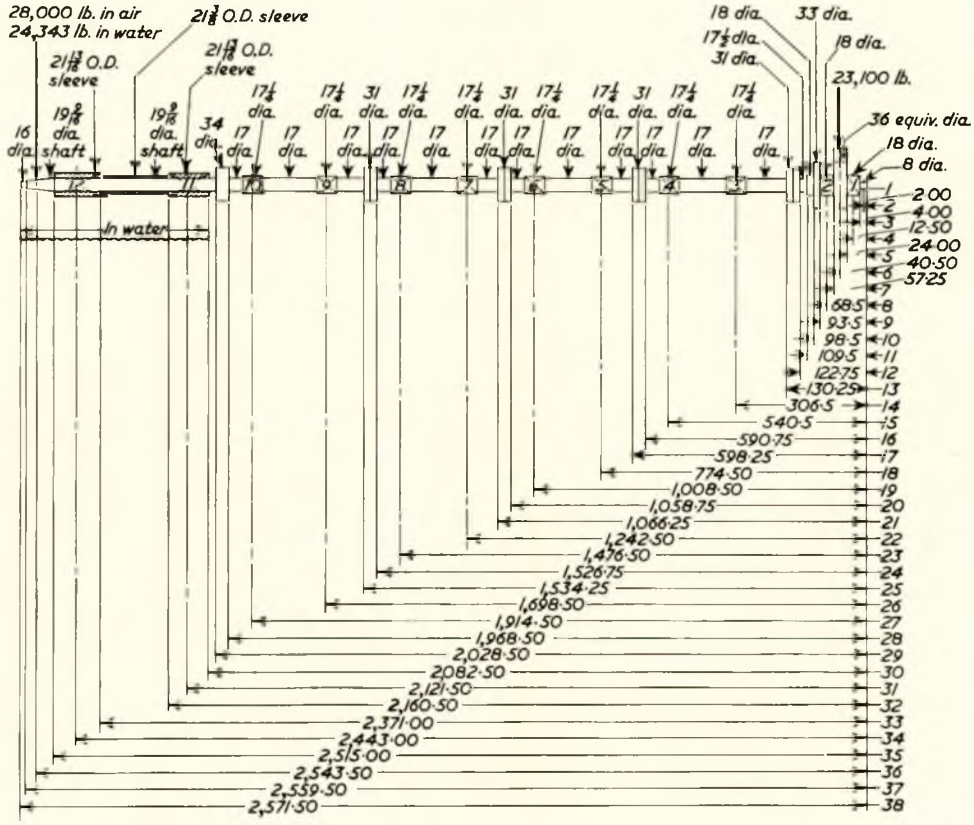
After the full power sea trials and the long distance maiden voyage, the tooth contact between the second-reduction pinions and the bull gear appeared as shown in Fig. 22. It will be noted from this chart that the tooth contact is *heavy* on the *aft end,* and only partially across each helix of the low pressure, second-reduction pinion, and practically all across, but slightly *heavy* on the *forward end* of each helix on the high pressure second-reduction pinion. This agrees with the calculations for unequally loaded gear bearings, in that the low pressure pinion is the "worst-affected" one, and the heavier tooth loading is on the ends, as calculated when heaviest bearing reaction is on the aft bull gear bearing.

The bearing loads were measured at the first availability and showed the aft bull gear bearing was excessively overloaded, exactly as could be expected by reviewing the line-up procedure, and referring to Table IX, column 1. As yet, the unequal load condition on this bull gear has not been corrected, due to the ship being unavailable, but without a doubt, when and if it is possible to realign the line shaft so that nearly equal static vertical loads are obtained on bull gear bearings, the tooth contact will be distributed uniformly across the face width, as illustrated in Fig. 23.

SHAFT AND GEAR ALIGNMENT CONDITIONS ON A MIDSHIP ENGINE ROOM INSTALLATION

Studies were made on a relatively long line shaft arrangement on a twin-screw ship, having eight line shaft metal bearings, as shown in Fig. 24. Table XII, column 1, shows the bearing reactions with all bearings on a straight line. Since the calculating time is influenced appreciably by the number of bearings involved, it can be reduced by considering only the first few bearings when determining the influence of line shaft bearings and gear bearings on one another. Column 2 of Table XII, therefore, shows the straight line reactions obtained when only the first six bearings were used in the calculations. (The shaft was considered severed midway between bearings numbers 6 and 7.) It can be noted that the reactions are practically the same values as obtained when using all twelve bearings.

Table XIII shows the bearing reaction influence numbers obtained for each of these conditions, and it can be noted they also compare closely. For the twelve-bearing consideration, only the first seven bearings are shown. The plus and minus signs in the table denote how the reaction of any particular bearing changes due to raising another bearing. A plus sign denotes the reaction increases, while a minus sign indicates a decrease in reaction. The signs must be reversed when a bearing is lowered. For example, when number 1 bearing is raised 0·010in., its bearing reaction is increased $10\times622=6{,}220$lb., and the reaction load on number 2 bearing is decreased $10\times805=8{,}050$lb. If bearing number 1 were lowered 0·010in., its bearing reaction would decrease 6 220lb., and the reaction load on number 2 bearing would increase 8 050lb. As mentioned before, the bull gear and line shaft bearing reactions, with all bearings in line in an assumed cold condition, are shown in columns 1 and 2 of Table XII. Column 3 shows the resulting bearing reactions in the hot operating condition when bull gear bearing journals have



Port and starboard similar

All dimensions are in inches

Total weight of line shaft gear and shaft = 40,500 lb. Shaft only = 11,050 lb.
Estimated equivalent diameter of hub for lateral stiffness = 36-in. outside diameter.

FIG. 24—*Line shaft arrangement for midship engine room*

TABLE XII.—CALCULATED LINE SHAFT BEARING REACTIONS

Bearing number	1 All bearings in line Cold: using all 12 bearings	2 All bearings in line Cold: using only 6 bearings	3 No. 1 up 0·020 in. No. 2 up 0·020 in. Hot*	4 No. 3 up 0·055 in. Cold*	5 No. 1 up 0·020 in. No. 2 up 0·020 in. No. 3 up 0·055 in. Hot*	6 No. 3 up 0·075 in. No. 4 up 0·068 in. Cold*	7 No. 1 up 0·020 in. No. 2 up 0·020 in. No. 3 up 0·075 in. No. 4 up 0·068 in. Hot*
1	11,223	11,199	7,563	23,983	20,323	24,475	20,815
2	33,680	33,720	38,740	15,310	20,370	15,702	20,762
3	15,266	15,199	13,226	25,166	23,126	20.946	18,906
4	15,996	16,232	16,856	9,671	10,536	18,815	17,955
5	15,239	14,362	15,019	17,769	17,549	11,549	11,329
6	15,997	15,760	16,057	15,337	15 397	18,021	18,081
7	15,279		15,259	15,444	15,424	14,688	14,668
8	15,846						
9	13,439						
10	23,300						
11	15,899						
12	59,462						

*These results were obtained by using the bearing influence numbers in Table XIII.

TABLE XIII.—CALCULATED BEARING REACTION INFLUENCE NUMBERS
(Pounds per 0·001-in. rise of bearings)

Bearing number	1	2	3	4	5	6	7
			With all 12 bearings taken into consideration				
1	+622	−805	+232	−61	+17	−4	+1
2	−805	+1,058	−334	+104	−28	+7	−2
3	+232	−334	+180	−115	+46	−12	+3
4	−61	+104	−115	+143	−105	+43	−12
5	+17	−28	+46	−105	+139	−104	+44
6	−4	+7	−12	+43	−104	+140	−104
7	+1	−2	+3	−12	+44	−104	+140
			With only first 6 bearings taken into consideration				
1	+622	−805	+232	−61	+15	−3	
2	−805	+1,058	−334	+103	−26	+4	
3	+232	−334	+180	−114	+43	−70	
4	−61	+103	−114	+140	−93	+25	
5	+15	−26	+43	−93	+95	−35	
6	−3	+4	−70	+25	−35	+15	

risen 0·020in., and it can be noted that bearing number 1 reaction has decreased, while number 2 reaction has increased, making the gear bearings still more unequally loaded.

Columns 4 and 5 show that the gear bearings can be equally loaded in the hot operating condition by raising number 3 line shaft bearing 0·055in.; and columns 6 and 7 show the same effect being accomplished by raising two line shaft bearings, number 3 raised 0·075in. and number 4 raised 0·068in. In the latter, the effect is to "fair" the shaft and reduce the maximum line shaft bearing reaction, which occurred on number 3 bearing.

Table XIV shows the results of a study to determine the effects of varying torque on tooth mismatch across one helix with given amounts of unequal loading on bull gear bearings for this particular gear unit. The table also shows the static downward loads assumed on forward and aft bearings and the calculated centre distances and out-of-plane conditions of mating pinion and gear journals.

Fig. 25(a) is a curve of unequal load *versus* mismatch across one helix for 100 per cent torque, plotted from data for Cases F and G for each pinion in Table XIV. Note that the high pressure pinion 2 is the "worst-affected" and therefore will be used to determine the allowable difference in static loading on bull gear bearings when at hot, operating conditions. For this particular gear unit the authors would allow a mismatch of 0·0003in., due to unequal loading; and from the curve it can be seen that this limits the unequal loading at 6,200lb.

Fig. 25(b) is also plotted from the data in Table XIV, using Cases B, E, F, and I for each pinion, showing the mismatch for 20,000lb., unequal loading *versus* per cent rated torque. It can be seen that the mismatch across one helix is affected by the per cent rated torque, and the maximum misalignment occurs at approximately 70 per cent for the worst affected pinion, and at approximately 100 per cent torque for the least affected. It is interesting to note that on the latter pinion, from 0 to about 30 per cent torque, the opposite end of the helix is open.

The effect of torque on misalignment makes it difficult to determine, by observing tooth contact marking, whether or not an excessive amount of tooth misalignment actually exists at full power. For example, if a running tooth contact check is made by applying a tooth marking medium such as Dykem on the pinion teeth, and then the gear is operated for long periods of time at light torque loads, the actual tooth contact is more uniform, and the tooth contact indication on the dye may be rubbed off uniformly across the entire face width. However, when the gear is then operated at higher torque loads up to maximum power, the gear may assume a crossed-

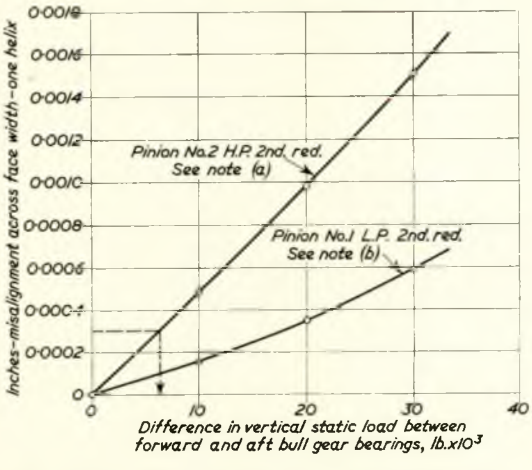


FIG. 25(a)—*Curves of tooth contact mismatch* versus *unequal gear bearing load*

Note: Heavier load can be forward or aft gear bearing.

a) Forward end of helix is open when heavier load is on aft bearing; aft end is open when heavier load is on forward bearing.

b) Aft end of helix is open when heavier load is on aft bearing; forward end of helix is open when heavier load is on forward bearing.

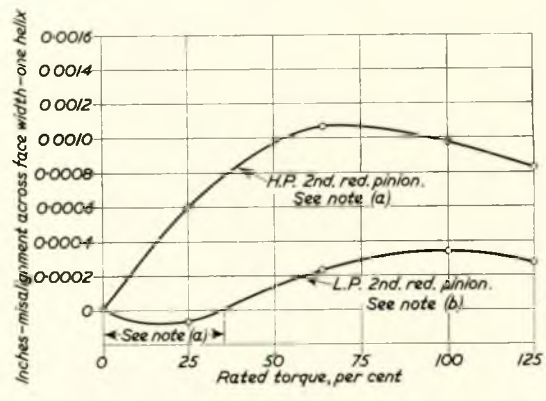


FIG. 25(b)—*Curves of tooth contact mismatch* versus *per cent rated torque at 20,000-lb. unequal gear bearing load*

Note: Calculations based on 20,000-lb. difference in load on bull gear bearings.

a) Forward end of helix is open when heavier load is on aft bearing; aft end of helix is open when heavier load is on forward bearing.

b) Aft end of helix is open when heavier load is on aft bearing. Forward end of helix is open when heavier load is on forward bearing.

TABLE XIV.—SUMMARY OF MISALIGNMENT CALCULATIONS

Case	% rated torque	Bull gear bearing loading, lb.			Centre distance between pinions and gear			Out of plane with bearing bores			Effect on backlash due to difference in in CD*	Mismatch across one helix* normal
		Difference	Forward bearing	Aft bearing	Forward end	Aft end	Difference in CD*	Forward end	Aft end	Difference*		
Low pressure second-reduction pinion (ahead rotation)												
A	25	10,000	17,000	27,000	79·67233	79·67314	−0·00081	0·01499	0·01457	0·00042	−0·00029	0·00004
B	25	20,000	12,000	32,000	79·67153	79·67337	−0·00184	0·01533	0·01444	0·00089	−0·00066	0·00007
C	64	10,000	17,000	27,000	79·66938	79·67105	−0·00168	0·01576	0·01548	0·00028	−0·00060	−0·00009
D	64	15,000	14,500	29,500	79·66878	79·67820	−0·00256	0·01578	0·01541	0·00037	−0·00093	−0·00016
E	64	20,000	12,000	32,000	79·66808	79·67160	−0·00352	0·01577	0·01533	0·00045	−0·00127	−0·00024
F	100	20,000	12,000	32,000	79·66712	79·67042	−0·00331	0·01561	0·01560	0·00001	−0·00120	−0·00034
G	100	30,000	7,000	37,000	79·66582	79·67092	−0·00510	0·01530	0·01550	−0·00020	−0·00185	−0·00058
H	123	10,000	17,000	27,000	79·66834	79·66972	−0·00137	0·01571	0·01568	0·00003	−0·00050	−0·00013
I	123	20,000	12,000	32,000	79·66746	79·67025	−0·00279	0·01563	0·01561	0·00002	−0·00101	−0·00028
Low pressure second-reduction pinion (astern rotation)												
J	100	10,000	17,000	27,000	79·67770	79·67857	−0·00087	0·01358	0·00987	0·00371	−0·00031	0·00097
K	100	15,000	14,500	29,500	79·67714	79·67856	−0·00143	0·01463	0·00915	0·00548	−0·00051	0·00142
L	100	20,000	12,000	32,000	79·67644	79·67852	−0·00208	0·01565	0·00852	0·00714	−0·00075	0·00182
High pressure second-reduction pinion (ahead rotation)												
A	25	10,000	17,000	27,000	79·67559	79·67507	0·00052	0·00438	0·00363	0·00075	0·00018	0·00027
B	25	20,000	12,000	32,000	79·67603	79·67490	0·00113	0·00512	0·00343	0·00170	0·00041	0·00060
C	64	10,000	17,000	27,000	79·67809	79·67760	0·00049	0·00745	0·00583	0·00162	0·00018	0·00051
D	64	15,000	14,500	29,500	79·67820	79·67747	0·00072	0·00804	0·00555	0·00249	0·00026	0·00078
E	64	20,000	12,000	32,000	79·67828	79·67737	0·00091	0·00873	0·00531	0·00342	0·00032	0·00107
F	100	20,000	12,000	32,000	79·67851	79·67806	0·00045	0·00978	0·00651	0·00327	0·00016	0·00098
G	100	30,000	7,000	37,000	79·67839	79·67789	0·00049	0·01111	0·00603	0·00508	0·00018	0·00150
H	123	10,000	17,000	27,000	79·67851	79·67830	0·00021	0·00860	0·00724	0·00136	0·00008	0·00041
I	123	20,000	12,000	32,000	79·67856	79·67818	0·00039	0·00949	0·00672	0·00277	0·00014	0·00083

*Indicates aft end open; otherwise forward end is open.

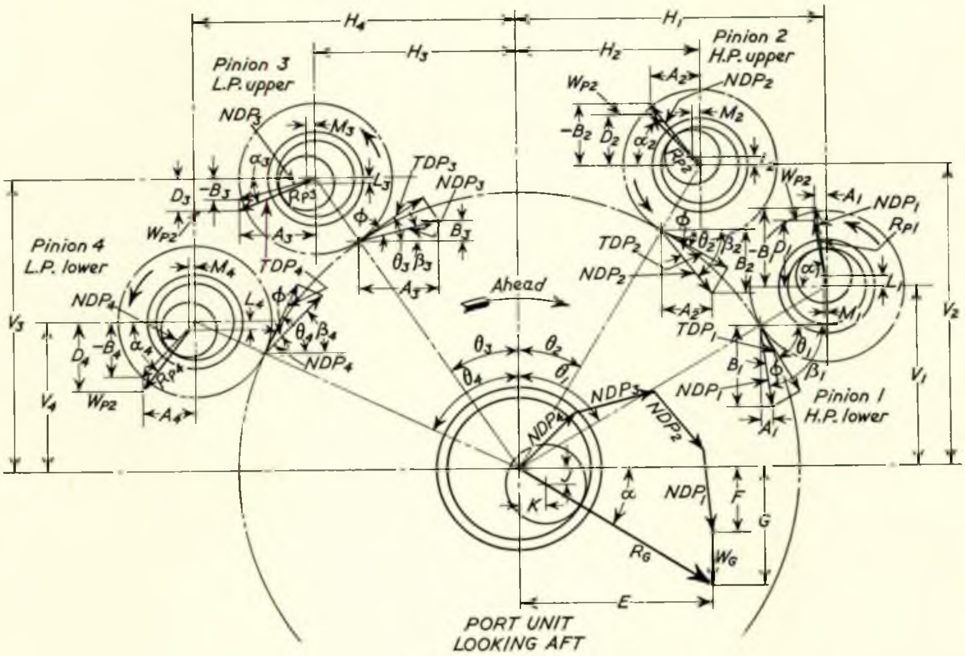


FIG. 26—*Bearing reaction diagram on double-locked train gear at 100 per cent torque*

Given data	
Centre distance (C), in.	73·16665
Pressure angle (ϕ), deg.	20
Pitch diameter, pinions (Dp), in.	16·00
$\frac{1}{2}$ diametral oil clearance, pinion (Bp), in.	0·0075
Tangential driving pressure (TDP), per pinion, lb.	46,026
Normal driving pressure (NDP), lb.	48,980
Weight, pinion (Wp), lb.	− 1,442
Weight, gear (W_G), lb.	−32,835
$\frac{1}{2}$ diametral oil clearance, gear (B_G), in.	0·0105

		Pinion 1	Pinion 2	Pinion 3	Pinion 4
1	H, in.	−63·765	−38·173	41·509	66·750
2	V, in.	35·880	62·419	60·252	29·964
3	$\theta=\tan^{-1}\frac{H}{V}$, deg.	−60·6338	−31·4482	34·5636	65·8253
4	$\beta=\theta-\phi$, deg.	−80·6338	−51·4482	14·5636	45·8253
5	$A=NDP\cos\beta$, lb.	7,971	30,525	47,406	34,132
6	$B=NDP\sin\beta$, lb.	− 48,327	− 38,305	12,316	35,129
7	$D=Wp-B$, lb.	46,885	36,863	− 13,758	− 36,571
8	$\alpha=\tan^{-1}\frac{D}{A}$, deg.	80·3513	50·3730	−16·1836	−46·9757
9	$R_p=A/\cos\alpha$, lb.	47,558	47,861	49,362	50,024
10	$L=B_p\sin\alpha$, in.	0·0074	0·0058	− 0·0021	− 0·0055
11	$M=B_p\cos\alpha$, in.	0·0013	0·0048	0·0072	0·0051

		Gear
12	$E=\Sigma A$, lb.	120,034
13	$F=\Sigma B$ lb.	−39,187
14	$G=F+W_G$, lb.	−72,022
15	$\gamma=\tan^{-1}\frac{G}{E}$, deg.	−30·9643
16	$R_G=E/\cos\gamma$, lb.	139,983
17	$J=B_G\sin\gamma$, in.	0·0054
18	$K=B_G\cos\gamma$, in.	0·0090

NOTE: This diagram shows total resultant forces on teeth and on two bearings of each rotor at 100 per cent torque; and assumes that the weight of each respective rotor splits 50–50 between its bearings.

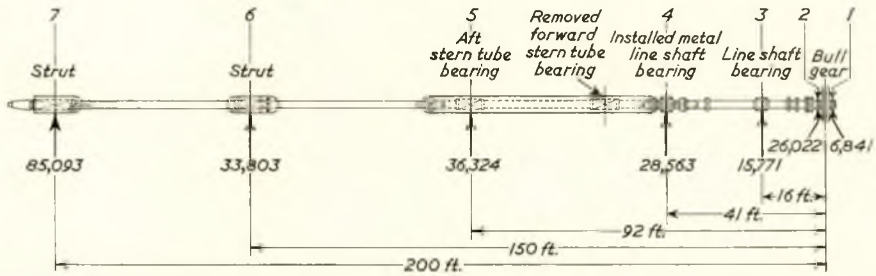


FIG. 27—*Line shaft and stern tube arrangement of bearings on twin-screw navy ship port unit*

axes condition if there is a large difference in loading on the two bull gear bearings, and the actual tooth contact may only be 50 to 75 per cent across the face width. Then, with the dye already rubbed off at the low torque runs, it would be difficult to determine the actual contact at full power. It is advisable, therefore, to apply a copper coating to a band of teeth, in addition to the band of teeth with dye, since the copper is not so easily rubbed off at the low torque loads.

To ensure proper installation and prevent internal misalignment of the gear, the following shaft alignment data and specifications were furnished to the shipyard by the gear manufacturer, and apply to both the port and starboard units, which are similar :

1) With the gear foundation, gear casings, and line shaft bearing pedestals at operating temperature and with the bull gear journals in their full load operating positions, the difference between the vertical downward loadings on the forward and aft bull gear bearings should not exceed 6,200lb.

2) The heavier load can be either the forward or aft bull gear bearing.

3) When the heavier load is on the aft bull gear bearing the forward end of the helices will be open on the high pressure, second-reduction pinion in the ahead rotation. The reverse or aft ends of the helices will be open when the larger load is on the forward bull gear bearing.

4) On the low pressure, second-reduction pinion ahead rotation, the aft end of the helices will only be open when the heavier load is on the aft bull gear bearing. The reverse will be true when the greater load is on the forward bull gear bearing.

5) The line shaft should be aligned to the bull gear shaft in the athwartship direction with no significant forces (less than 1,000lb.) exerted in the horizontal plane on the bull gear bearings.

6) With an oil clearance of 0·020in. between the bull gear shaft journals and bores of the bearings, the centre of the bull gear shaft will rise vertically approximately 0·0025in. and move athwartship about 0·0065in. at full power operating condition.

7) Owing to thermal expansion of the gear foundation and lower gear housing, it is estimated that the centre line of the bull gear shaft will rise vertically 0·020in. from the cold to the hot operating condition.

8) When bolted to the line shaft, the bending stress in the bull gear shaft should not exceed 15,000lb. per sq. in.

9) The maximum bearing load reaction on either bull gear bearing, in the static cold or hot condition, should not exceed 43,200lb. in order to limit the unit load to 150lb. per sq. in.

SHAFT ALIGNMENT RECOMMENDATIONS ON A TWIN-SCREW NAVY VESSEL

Shaft alignment studies were recently completed on a high powered twin-screw navy ship using a double-locked train gear with four second-reduction pinions and an integral main thrust bearing on the aft end of the propulsion gear. The arrangement of the four pinions around the bull gear is shown in Fig. 26. The position of the pinion and gear journals in their bearings at 100 per cent torque, with equal loads on the two bull gear bearings, are also shown in this figure.

The final arrangement of the propeller shaft and bull gear is shown in Fig. 27 for the port shaft. In the original arrangement there were two water lubricated stern tube bearings and one oil lubricated metal line shaft bearing. With this arrangement a satisfactory gear and shafting alignment condition could not be obtained. The reason for this is the large allowable wear-down for the water lubricated bearings, which in this case was 0·223in. With a wear-down of only 0·018in. on the forward stern tube bearing the aft bull gear bearing would be completely unloaded in the hot operating, full power condition. Under this condition there would be serious misalignment between the pinions and the bull gear due to skewing of the bull gear journals in the bearings. To eliminate such a possible condition the forward stern tube bearing was changed to an oil lubricated metal bearing, as shown in Fig. 27. The figures adjacent to each shaft bearing are the bearing reaction loads with all bearings on a straight line and correspond to the figures listed at the top of Table XV. This table also lists the bearing influence numbers per 0·001-in. vertical movement of each bearing on the port shaft.

Owing to the short two-bearing line shaft and the close proximity of the forward line shaft bearing to the aft bull gear bearing, the bull gear shaft alignment to the line shaft is quite sensitive to any change. It will be noted from Table XV that the bearing reaction load influence per 0·001-in. vertical

TABLE XV.—BEARING REACTIONS AND INFLUENCE NUMBERS

Bearing number	1 Forward gear	2 Aft gear	3 1st line shaft	4 2nd line shaft	5 Stern tube bearing	6 Forward strut bearing	7 Aft strut bearing
	Reaction loads with all bearings in straight line						
	6,841	26,022	15,771	28,563	36,324	33,803	85,093
	Bearing reaction influence numbers (lb. per 0·001-in. bearing rise)						
1	1,781	−2,309	594	−73	10	−2	0·5
2	−2,309	3,043	−856	135	−18	4	−0·9
3	594	−856	360	−116	23	−5	1
4	−73	135	−116	72	−26	9	−2
5	10	−18	22	−26	21	−14	4
6	−2	4	−6	9	−14	14	−6
7	0·5	−0·9	1	−2	4	−6	3

TABLE XVI.—BEARING REACTIONS WITH VARIOUS SETTINGS OF LINE AND GEAR SHAFT BEARINGS

Column no. / Bearing	1 Cold No. 1 down 0·034 in. No. 2 down 0·034 in.	2 Hot (A)* No. 1 down 0·015 in. No. 2 down 0·015 in.	3 Hot (B)* No. 1 down 0·010 in. No. 2 down 0·010 in.	4 Hot (C)* No. 1 down 0·004 in. No. 2 down 0·004 in.	5 Cold No. 1 down 0·034 in. No. 2 down 0·034 in. No. 3 up 0·005 in.	6 Hot (B)* No. 1 down 0·010 in. No. 2 down 0·010 in. No. 3 up 0·005 in.	7 Cold No. 1 down 0·034 in. No. 2 down 0·034 in. No. 3 up 0·010 in.	8 Hot (C)* No. 1 down 0·004 in. No. 2 down 0·004 in. No. 3 down 0·010 in.
No. 1 Forward bull gear bearing	24,793	14,761	12,121	8,953	27,763	15,091	30,733	14,893
No. 2 Aft bull gear bearing	1,066	15,012	18,682	23,086	− 3,214	14,402	− 7,494	14,526
No. 3 First line shaft bearing	24,679	19,701	18,391	16,819	26,479	20,191	28,279	20,419
No. 4 Second line shaft bearing	26,455	27,633	27,943	28,315	25,875	27,363	25,295	27,155

**Assumed Hot Conditions*
(A) Bull gear journals rise 0·019 in.
(B) Bull gear journals rise 0·024 in.
(C) Bull gear journals rise 0·030 in.

change in position of bearings numbers 1 and 2 is quite large. Any small change in the calculated or estimated vertical expansion of the gear foundation and housing would make an appreciable difference in load between the forward and aft bull gear bearings numbers 1 and 2. Any large difference in load on the two bull gear bearings would result in bad internal misalignment between the bull gear and pinions.

The bearing load reactions for numbers 1, 2, 3 and 4 bearings, under various conditions of elevations of the bearings, are shown in Table XVI. (Bearings numbers 5, 6 and 7 are not included, since their influence on gear bearings is considered negligible.) Bearing reactions in column 1, in the cold condition, are with bearings numbers 1 and 2 set 0·034in. lower than the bearings numbers 3 and 4. The bearing load reactions in column 2 are for the hot operating condition, assuming that bull gear shaft journals will rise 0·019in. from the cold setting. Under this condition the difference in loading between the two bull gear bearings is negligible, therefore this would be satisfactory. If the bull gear shaft journals rose 0·024in. between the cold and hot operating conditions, the difference in load between the bull gear bearings would be 6,561lb., as shown in column 3; and this condition is also satisfactory. However, if the bull gear journals rose 0·030in. going from the cold to the hot operating condition, the difference in bull gear bearing loading would be 14,133lb., as shown in column 4. This is an unsatisfactory condition, as it results in a mismatch or opening between pinion 4 and the gear of 0·00042in. on the forward end of each helix. This is shown on curves in Fig. 28.

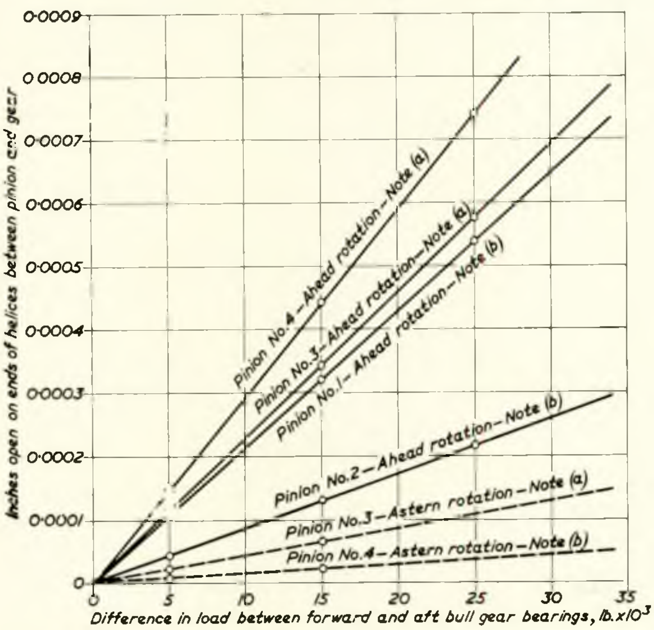


FIG. 28—*Curves of tooth face mismatch on four pinions* versus *unequal load on bull gear bearings*

Note: Heaviest load can be on either end.
a) Forward ends of helices are open when heaviest load is on aft bearing; aft ends of helices are open when heaviest load is on forward bearing.
b) Aft ends of helices are open when heaviest load is on aft bearing; forward ends of helices are open when heaviest load is on forward bearing.

In view of the foregoing conditions, it was recommended that the bull gear shaft centre line be set between 0·019 and 0·024in. lower than line shaft centre in the cold condition. Since the calculated thermal expansion of the gear foundation and lower gear housing was approximately 0·030in. from the cold to the hot operating condition it was also suggested that the bearing reactions on bearings numbers 1 and 2 be weighed in both the cold and hot conditions. If the weighing checks indicated unequal loads in the order of 14,000lb. as shown in column 4, then it would be necessary to raise number 3 bearing 0·005in. in the cold condition as indicated in column 5, Table XVI. Although there would be a negative load of 3,214lb. on the aft bull gear bearing in the cold condition, this is not considered too serious. Since the bearing reaction load on the two bull gear bearings would be practically equal at the full power hot operating condition, the small negative load on the aft bull gear bearing at very light loads will do no harm.

ALIGNMENT OF TURBINES TO GEAR FOR A TYPICAL MARINE PROPULSION UNIT

This description of alignment procedures is applicable to a cross-compound, geared turbine marine propulsion unit. In such an installation, the driving turbines are connected to the first-reduction pinions of the propulsion gear by dental-type flexible couplings. Of vital concern is the alignment of these couplings, and consequently the alignment of the turbine rotors with the first-reduction pinions in order to reduce to a minimum the amount of coupling wear that will occur when they are transmitting maximum load over a continuous period. To better understand the recommended turbine alignment procedure, a description of the installation arrangement, the relative expansions between the various parts of the installation, the analyses of these expansions, and finally the construction of layouts to arrive at recommended cold settings for use when installing the equipment, is briefly outlined.

Description of Installation

The ship's foundation in way of the reduction gear has the usual hollowed-out well to receive the lower extension of the bull gear when it is supported in the low speed journal bearings. The lower extension of this oiltight well also serves as the hot oil sump. A seal is provided between the foundation and the lower gear casing flange after the casing is chocked to align the gear to the line shaft.

Fig. 29 shows a typical foundation for an installation similar to the one being described. It will be noted that the high pressure turbine is supported by a beam girder construction running fore and aft. The forward end of the beam is supported by a transverse wall structure, the other side of which serves as a support for the main condenser. The aft end of the high pressure turbine beam is supported by a bracket which is in turn anchored to the middle gear casing. In this arrangement, the aft end of the high pressure beam will expand vertically more than the forward end since it will rise with the gear casing which is exposed to higher operating temperatures than the transverse wall structure at the forward end.

There is another transverse structure supporting the aft end of the condenser as shown in the figure. This structure is tied in fore and aft with the middle gear casing but is free to expand vertically, independent of the gear. There is sufficient athwartship flexibility in these fore and aft braces to allow expansion of the gear and condenser relative to one another.

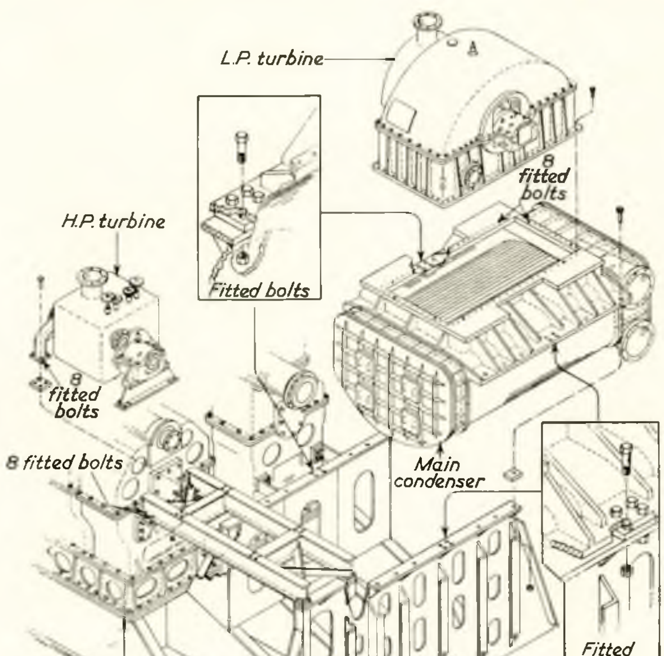


FIG. 29—*Typical foundation structure for propulsion turbine gear*

The low pressure turbine rests directly on the main condenser; and is supported by it down through the two transverse structures to the foundation.

Description of Relative Expansions

The area of the foundation under the bull gear is exposed to the hot oil that is draining back to the sump from the first- and second-reduction gear meshes and bearings. Consequently the foundation under the second-reduction elements will expand vertically more than under the first-reduction elements.

This condition is shown in Fig. 30, which presents a representative distribution of wall temperatures for a typical marine propulsion unit with oil inlet temperature to gear 120 deg. F. These temperatures have been determined by actual measurement during factory tests. They will vary, however, with the design of the gear housing and will not in many cases be the same as shown here. The form in Fig. 30 provides a convenient method of arranging and grouping all of the applicable temperatures and dimensions that are required to derive an alignment procedure for any installation. Note that the datum for all temperatures and dimensions is taken as the tank top.

The axial oil leakage past the journal bearings supporting all of the gear rotors, combined with the oil spray from mesh lubrication, washes down the internal walls of the gear casing. The walls closest to the outside of the gear casing give off some of their heat by radiation to the engine room, and consequently " run cooler " than the inner walls. This condition can be seen by referring to Fig. 30. The net result is that the aft bearing of the first-reduction pinions will expand, with reference to the tank top datum, a larger amount than the forward bearing when the gear unit and foundation are heated from a given assembly room temperature to a hot operating condition. This will be covered in more detail later.

The vertical expansion of the turbine's supporting structure in this case is such that the low pressure turbine centre line rises equally at each end. The high pressure turbine tilts, however, since the forward supporting structure expands less than the aft end, which rises with the middle gear casing. For each turbine, the expansion of the supporting feet are calculated for full ahead power, since this is the condition at which the unit will operate over the major part of its life.

In the athwartship direction, the high pressure turbine is again pulled outboard at the aft end along with the gear casing. The low pressure turbine is not so constrained, however, owing to the lateral flexibility of the fore and aft braces between the supporting structure and the middle gear casing.

Calculations

The basic assumptions made in calculating the relative expansions in the unit are as follows:

1) Engine room assembly temperature = 80 deg. F.

2) Oil inlet temperature = 120 deg. F.

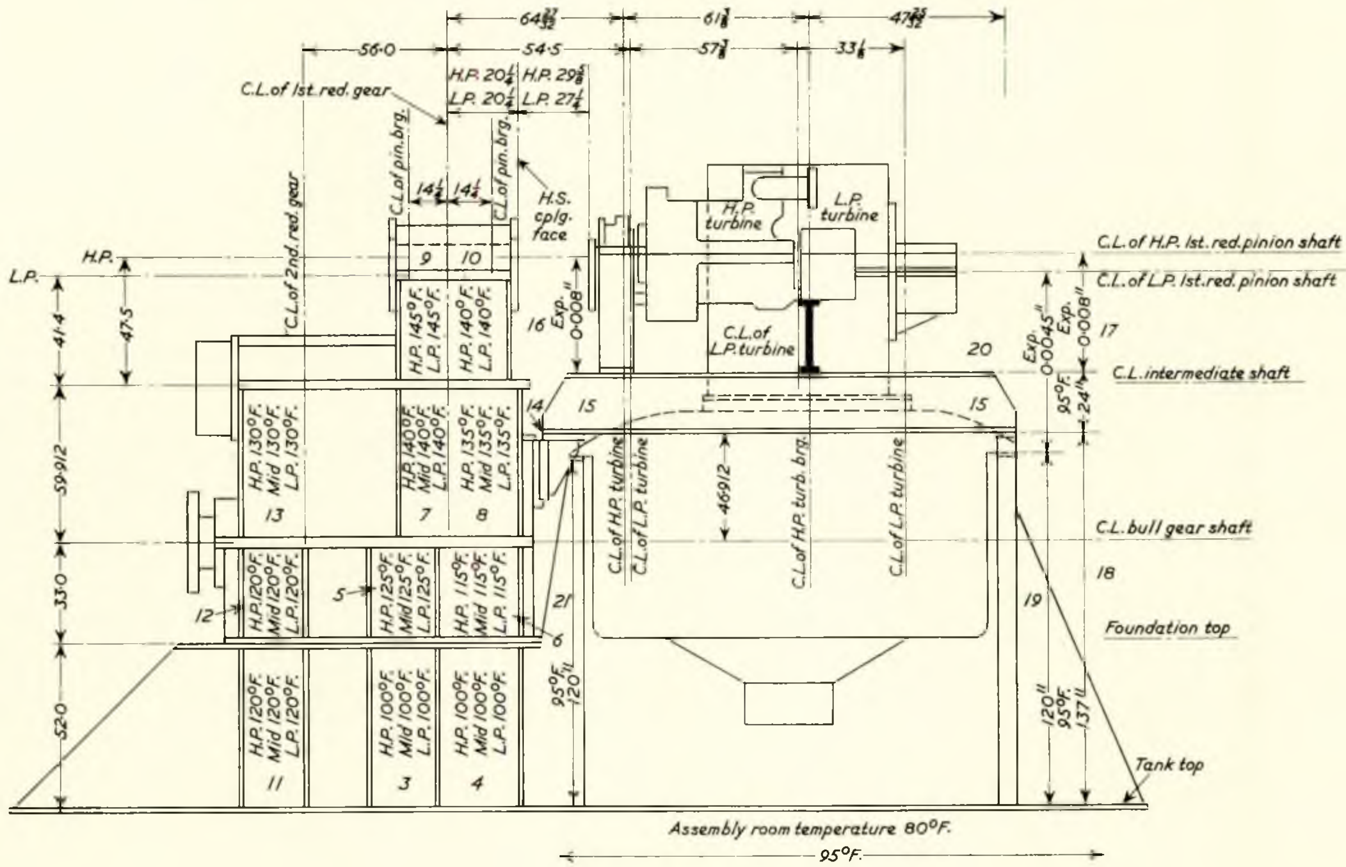
3) Coefficient of linear expansion for steel = $6 \cdot 3 \times 10^{-6}$/deg. F. = α.

4) Foundation expansions are calculated, using the ship's inner bottom or tank top as a reference.

5) When making alignment checks, the first-reduction pinions are strapped or held down and secured centrally sideways in their bearing clearances.

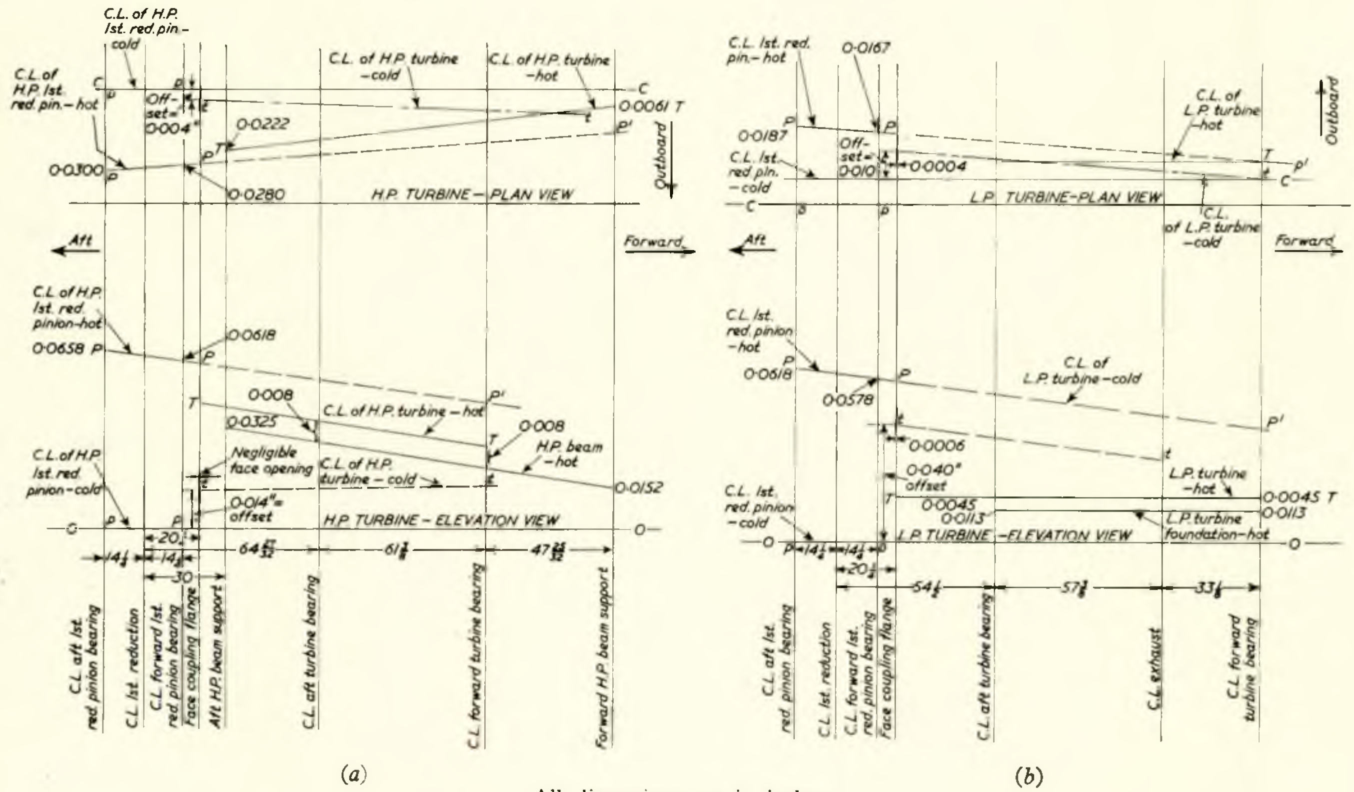
A sample calculation form is shown in Table XVII, on which the values of lengths and temperatures shown in Fig. 30 are used to arrive at actual expansions. Note that with the alignment made as just described, with the pinions strapped down in their bearings, allowance must be made for the reaction positions of the pinions when they are transmitting full ahead torque. Such allowance is considered in step (i) for the first-reduction pinions and in step (iv) for the bull gear.

It is preferable when considering the bearing reactions to choose a view which shows the pinions as the installation man will see them when he follows the prescribed alignment procedure. The usual practice is to take and record readings



All dimensions are in inches

FIG. 30—*Wall temperatures of a typical propulsion gear installation*



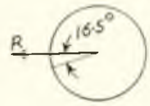
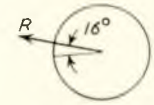
All dimensions are in inches

FIG. 31—(*a*) *High pressure and* (*b*) *low pressure turbine support spans layouts*

TABLE XVII.—SAMPLE CALCULATIONS FOR VERTICAL AND HORIZONTAL WALL EXPANSIONS

(i) *Bearing reaction compensation, single unit, looking aft.*

Nominal diametral bearing oil clearances
H.P. = 0·012
L.P. = 0·0115

	High pressure pinion		*Low pressure pinion*
			
Vertical rise	0·012/2 + 0·012/2 sin 16·5° 0·006 + 0·0017 = 0·0077	Vertical rise ..	0·0115/2 + 0·0115/2 sin 16° 0·0058 + 0·0016 = 0·0074
Athwartship motion	0·012/2 cos 16·5° = 0·0058 outboard	Athwartship motion	0·0115/2 cos 16° = 0·0055 inboard

(ii) *Vertical expansions – gear*

	High pressure	*Low pressure*
(*a*) *Foundation* (52·0)(α)(100 – 80) = 0·0066 in.		
(*b*) *After pinion bearing*		
1. Lower casing ..	(33·0)α(125 – 80) = 0·0094	(33·0)α(125 – 80) = 0·0094
2. Middle casing ..	(59·912)α(140 – 80) = 0·0227	(59·912)α(140 – 80) = 0·0227
3. Upper casing ..	(47·5)α(145 – 80) = 0·0194	(41·4)α(140 – 80) = 0·0157
4. Bearing reaction ..	Step (i) = 0·0077	Step (i) = 0·0074
5. Foundation	Step (ii-*a*) = 0·0066	Step (ii-*a*) = 0·0066
	0·0658	0·0618
(*c*) *Forward pinion bearing*		
1. Lower casing ..	(33·0)α(115 – 80) = 0·0073	(33·0)α(115 – 80) = 0·0073
2. Middle casing ..	(59·912)α(135 – 80) = 0·0208	(59·912)α(135 – 80) = 0·0208
3. Upper casing ..	(47·5)α(145 – 80) = 0·0194	(41·4)α(140 – 80) = 0·0157
4. Bearing reaction ..	Step (i) = 0·0077	Step (i) = 0·0074
5. Foundation	Step (ii-*a*) = 0·0066	Step (ii-*a*) = 0·0066
	0·0618	0·0578

(iii) *Athwartship expansions – gear*

(*a*) *After pinion bearing*		
1. Casing expansion ..	(64·0)α(140 – 80) = 0·0242 outboard	(64·0)α(140 – 80) = 0·0242 outboard
2. Bearing reaction ..	Step (i) = 0·0058 outboard	Step (i) = 0·0055 inboard
	0·0300 outboard	0·0187 outboard
(*b*) *Forward pinion bearing*		
1. Casing expansion ..	(64·0)α(135 – 80) = 0·0222 outboard	(64·0)α(135 – 80) = 0·0222 outboard
2. Bearing reaction ..	Step (i) = 0·0058 outboard	Step (i) = 0·0055 inboard
	0·0280 outboard	0·0167 outboard

(iv) *Expansions of bull gear, single unit, looking aft*
Nominal diametral clearance = 0·025

Vertical rise	0·025/2 – 0·025/2 sin 53 0·0125 – 0·010 = 0·0025	Foundation ..	(52·0)α(120 – 80) = 0·0131
		Lower casing ..	(33·0)α(120 – 80) = 0·0083
Athwartship motion	0·025/2 cos 53° = 0·0075	Bearing reaction ..	See left = 0·0025
			0·0239

(v) *Vertical expansions – turbines*

	High pressure	*Low pressure*
(*a*) *Foundation*		
1. Aft	0·0302 + (24)α(95 – 80) = 0·0325	(120)α(95 – 80) = 0·0113
2. Forward ..	(161·0)α(95 – 80) = 0·0152	(120)α(95 – 80) = 0·0113
(*b*) *Turbine expansions at full ahead power*		
1. Aft	0·008	0·0045
2. Forward	0·008	0·0045

(vi) *Athwartship expansion of turbines*

(*a*) Aft	(64·0)α(135 – 80) = 0·0222	(64·0)α(95 – 80) = 0·0061
(*b*) Forward	(64·0)α (95 – 80) = 0·0061	(64·0)α(95 – 80) = 0·0061

Note: The figures in the first bracket in each equation are in inches, in the second brackets deg. F.

looking aft from the turbines. In such a position, the high pressure pinion will ordinarily be on the left hand and it is so shown in Table XVII. In twin-screw vessels, the port unit will be reversed; that is, the high pressure pinion will be on the right hand.

The calculations are self-explanatory. It can be seen that the difference in total expansion between the after first-reduction pinion bearing (inner wall) and the forward first-reduction pinion bearing (wall closer to the outside) is 0·004 in. on both the high pressure and low pressure side (steps (ii-*b* and *c*)). For the same reason, the aft pinion bearings expand outboard 0·002in. more than the forward pinion bearings when considering the athwartship expansions (steps (iii-*a* and *b*)).

Construction and Description of Alignment Layouts

Figs. 31(a) and 31(b) are scale layouts of the spans between major supports in the installation. Fig. 31(a) shows the high pressure elements in side elevation and plan views, the fore and aft dimensions corresponding to those in Fig. 30. The left side of the figure represents the aft end of the installation and working forward there is the aft first-reduction pinion bearing, then the forward bearing, the coupling flange, the aft turbine bearing, and so on. A suitable scale to use for these layout is 0·1in. = 0·001in. vertical and 0·1 = 1·0in. horizontal.

With the pinion and the turbine centre lines in line and represented by line *O-O*, the total vertical expansion of the high pressure first-reduction aft pinion bearing (Table XVII, (ii-b)) is laid out to scale along the proper vertical line. Then the corresponding expansion of the forward pinion bearing is laid out and the two points are joined to form the line *P-P* extended to the face of the pinion coupling flange. The line

P-P then represents the position of the high pressure first-reduction pinion when the unit is at operating temperature. Next, the total vertical expansion of the aft high pressure turbine beam, where it is attached to the gear casing brackets, is laid out (Table XVII, (v-*a*1)) and the same is done at the forward high pressure turbine beam support (Table XVII, (v-*a*2)). The line connecting these two points represents the position of the high pressure turbine beam support at operating temperature. At the proper fore and aft locations, the vertical expansions of the turbine feet are laid out vertically (Table XVII, (v-*b*)). By connecting these two points a line *T-T*, representing the centre line of the high pressure turbine hot, is drawn. Graphically, the lines *P-P* and *T-T* show the relative misalignment between the pinion and turbine in the vertical direction, if they were set line-in-line on *O-O* when the unit was installed.

If the pinion centre line is extended to point p', it can be determined by measurement or calculation just what compensation must be made in aligning the unit cold so that the pinion and turbines are in line at full ahead power operation. With proper compensation the sliding action, due to misalignment in the dental-type flexible coupling connecting these elements, will be at a minimum, and the coupling wear likewise will be reduced as much as possible.

If the vertical distance between the lines *T-T* and *P-P-P'* at two points, *T-P* and *T-P'*, are laid out vertically from line *O-O* at the same axial points, they will establish two points, which when connected by a straight line, will form the line *t-t*. This then is the position the high pressure turbine centre line should have in relation to the pinion centre line when it is in position *p-p*, on line *O-O* at line up, so these parts will be in line in the hot operating condition.

To obtain alignment data, measure the vertical offset (0·014in.) between lines *p-p* and *t-t* at their intersection with coupling flange face, and in this case note that the turbine shaft centre line must be set higher than the pinion centre line. Also, measure or calculate the *angular* misalignment between the two lines and note its direction, that is, whether the flanges are open at the top or at the bottom of their peripheries, and ratio it to the sweeping diameter of the mating-coupling flange faces.

In the plan view at the top of Fig. 31(a) the centre line of the gear unit is at the top of the layout not shown and (line *C-C*) is the centre line of the h.p. pinion in the cold condition, from which athwartship expansions can be represented by measuring downward from this line. Using the calculated athwartship expansions from Table XVII, and making a layout as described for the vertical expansion, the required centre line offset and flange face opening in the horizontal plane for cold conditions can be obtained.

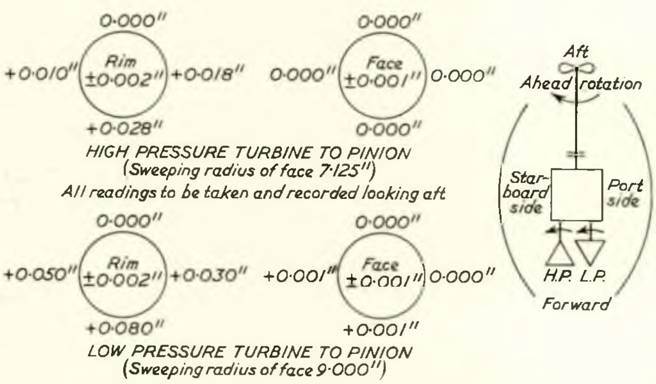


FIG. 32—*Indicator sweep measurements for turbine gear alignment*

Alignment of Turbines to Gears

This alignment is accomplished by using the h.p. and l.p. first-reduction pinions as arbors. These pinions will be strapped down in their bearings and will be located central sideways in their bearing clearances. The rim and face of these pinion coupling flanges will be swept by a dial indicator mounted on a suitable extension secured to the turbine coupling flanges. The sweeping procedure will then be accomplished by rotating the turbine shaft. The ambient temperature used as a basis for these settings is 80 deg. F.

Fig. 31(b) for the low pressure elements is constructed in exactly the same manner as described for Fig. 31(a).

Results and Recommendations for Alignment

Fig. 32 shows the alignment readings that are to be met in order to achieve the alignment worked out in Figs. 31(a) and 31(b). The note in the figure is self-explanatory, and describes how a dial indicator is used to sweep the flange face for angular setting and rim for offset setting of the strapped-down high speed pinions by rotating the turbine rotor. The note also states that the readings shown in the figure are to be measured and recorded looking aft.

In studying the cold offsets obtained in Fig 31(a), for example, it will be noted that the high pressure turbine shaft centre line must be set 0·014in. higher and 0·004in. outboard from the pinion centre line. Converting these to dial indicator rim sweep readings, obtained by setting the indicator dial at zero at the top of the pinion flange, will give a total of $2 \times 0{\cdot}014 =$ plus 0·028in. in the vertical direction at bottom; $0{\cdot}014 - 0{\cdot}004 =$ plus 0·010in. on the left hand; and $0{\cdot}014 + 0{\cdot}004 =$ plus 0·018in. on the right hand of the sweep circle. The actual offset in either direction is equal to one-half of the difference between rim readings diametrically opposed. As a check on these figures, the arithmetical sum across any two mutually perpendicular diameters should be equal.

Since in this case there was no angular misalignment between the flanges, all of the face-sweep readings are zero.

Presentation of the turbine-to-gear alignment by means of sweep readings, as in Fig. 32, is a simple method of expressing alignment between two parts. The desired alignment is accurately and relatively easily obtained when a suitable extension (no sag or compensated-for sag) is provided, and turbine rotated to obtain sweep reading data.

The authors' experience shows that when the foregoing procedure of calculating and establishing turbine-to-gear alignment is used, the coupling will be at maximum advantage in this regard, and will give good dependable high speed coupling operation.

CONCLUSIONS

To realize the optimum in performance from the modern, high precision, compact and lightweight ship propulsion gear, it is necessary that the shafting, pedestals and supports in the ship be designed properly, and the gear correctly installed, so that the complete line shaft arrangement is compatible under all conditions of operation. If not, certain bearings can be excessively loaded at one condition and then nearly unloaded at others, due to relatively minor changes in their position caused by such factors as the thermal expansion of the gear unit, wear-down of bearings, and so on.

On new construction it is possible to design the complete line shaft arrangement and alignment procedure to meet the specifications as presented herein regarding static bull gear loading. However, on existing designs, when bearing reactions are sensitive to changes in relative heights of bearings, a correct setting may be difficult to make.

It appears that most line shaft arrangements and alignment procedures used in the past result in an excessive static load on aft bull gear bearings. When the second-reduction teeth are worn into a pattern that satisfies a given alignment, or if pinion bearings have been scraped to obtain tooth contact distribution, then any changes of gear to line shaft alignment must be made with caution or an unsatisfactory tooth alignment may result.

Checking bearing reactions at prescribed intervals should be done as a preventative maintenance item; and the calibrated jack method, together with high speed computer programmes, make it possible to study and make necessary adjustments with a scientific approach more rapidly.

In the authors' opinion, the proposed method of analysis and alignment of the line shafting to the propulsion gear is a step in the right direction. It may provide an answer to some of the puzzling tooth contact and noise problems encountered on gears in the past. Operating experience and the data accumulated will indicate whether or not the present values on the limits of misalignment per helix, and hence the allowable unequal bull gear bearing loads, as herein stated, are correct or should be revised to smaller or greater values. The analysis of the propeller and line shafting, with its influence on the internal alignment of the bull gear and its meshing pinions, as presented herein, is theoretically correct; but the authors realize that training of personnel and experience with the use of the proposed method are necessary before a good understanding is obtained, and its full potential is realized.

The thermal expansions of turbines and gears must be considered when making an alignment of turbines to the first-reduction pinions, such as presented in this paper, in order to arrive at a good alignment at full power conditions. A good alignment is an important factor contributing to longer coupling life and elimination of some of the turbine vibration and flexible coupling difficulties experienced in the past.

ACKNOWLEDGEMENTS

The authors wish to acknowledge the co-operation and information made available to them by the Design Division of the Boston Naval Shipyard; particularly for the use of their IBM-650 Computer Program on Line Shaft Characteristics, which made possible the studies of the various line shaft arrangements used as examples in this paper. These examples were based on the approach to the problem presented at the January 1956 meeting of the New England Section of the Society of Naval Architects and Marine Engineers, in the paper, "The Alignment of Main Propulsion Shaft Bearings in Ships", by Lieut. R. E. Kosiba, U.S.N., J. J. Francis, and R. A. Woolacott, of the Boston Naval Shipyard. Without the studies and illustrative examples as set forth in that paper, this paper could not have been written.

The authors wish also to extend their thanks to other shipyards for their co-operation and information; namely, Electric Boat, Bethlehem Steel Quincy Shipyard, Newport News Shipbuilding and Dry Dock, Gibbs and Cox, and National Bulk Carriers, Inc.; and also to all their associates in the General Electric Company who have contributed their time and effort to the accomplishment of this paper, particularly Mrs. D. E. Bethune, for technical assistance and preparation of data.

Discussion Held at The Society of Naval Architects and Marine Engineers, New York, on 13th November 1959

Mr. J. J. Francis (Member) said the paper presented an excellent summary of the authors' accomplishments in the field of co-ordinated ships' propulsion equipment alignment. His discussion of the paper was primarily concerned with the relationship of inboard line shaft bearings to the co-ordinated alignment problem.

Current ship design practice tended towards installing at least one babbited line shaft bearing aft of the after bull gear bearing. For ships with long inboard line shafting installations, several babbited bearings were installed. When problems of heavily or lightly loaded inboard bearings occurred, local adjustments were made to relieve these situations. Frequently, time and money dictated the minimum corrective measures in order to return the ship to active service.

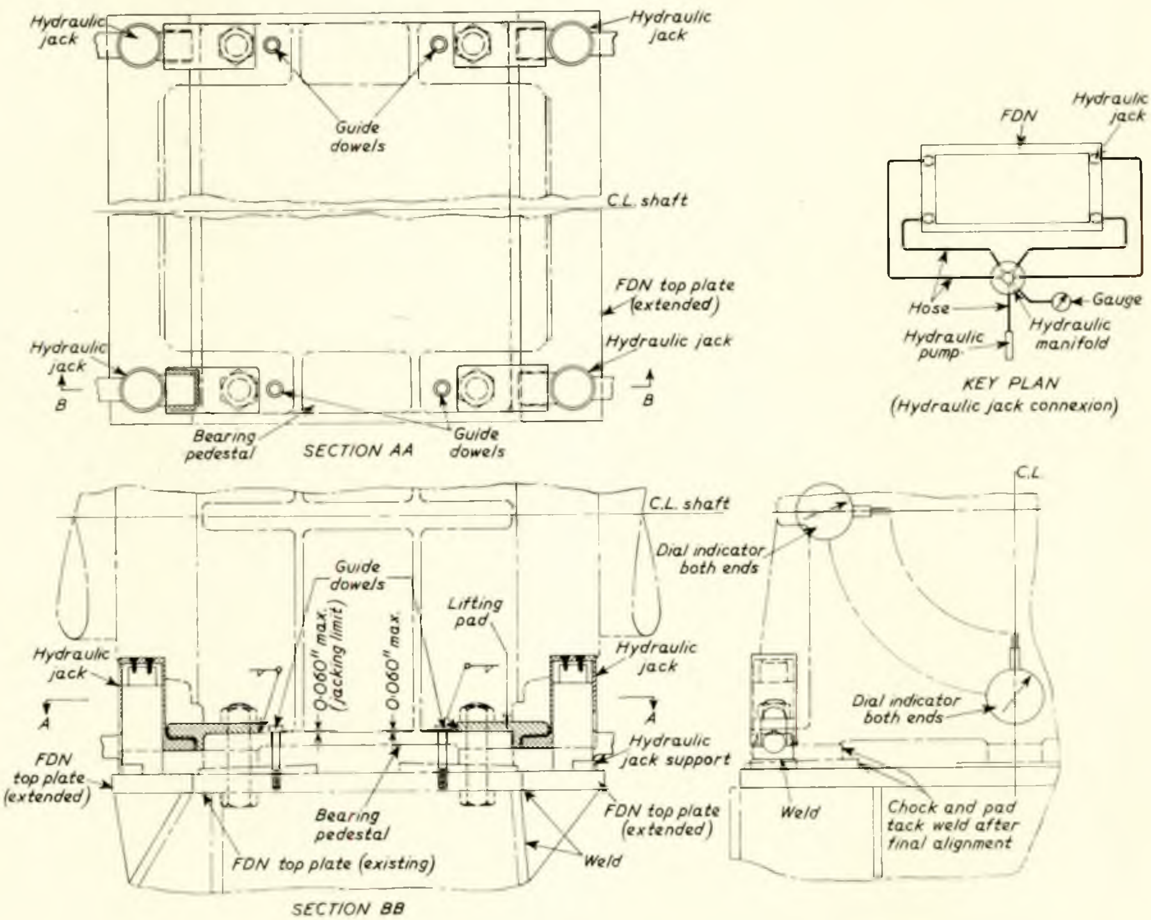


Fig. 33—*Portable load test jacks for steady bearing (hydraulic)*

Notes: Clamp dial indicators (4) on the forward and aft vertical and horizontal centre lines so as to measure motion between the bottom of the shaft and the foundation. Set the dial indicators to zero and remove fitted foundation bolts.

Elevate the jacks (4) *via* the hydraulic manifold so that the pedestal is just free of the foundation chocks.

Using the average reading of the dial indicators and the reading previously obtained by use of calibrated jacks, a jacking force can be arrived at which will be the actual shaft loading of the bearing.

(*These notes also apply to Figs. 34 and 35*)

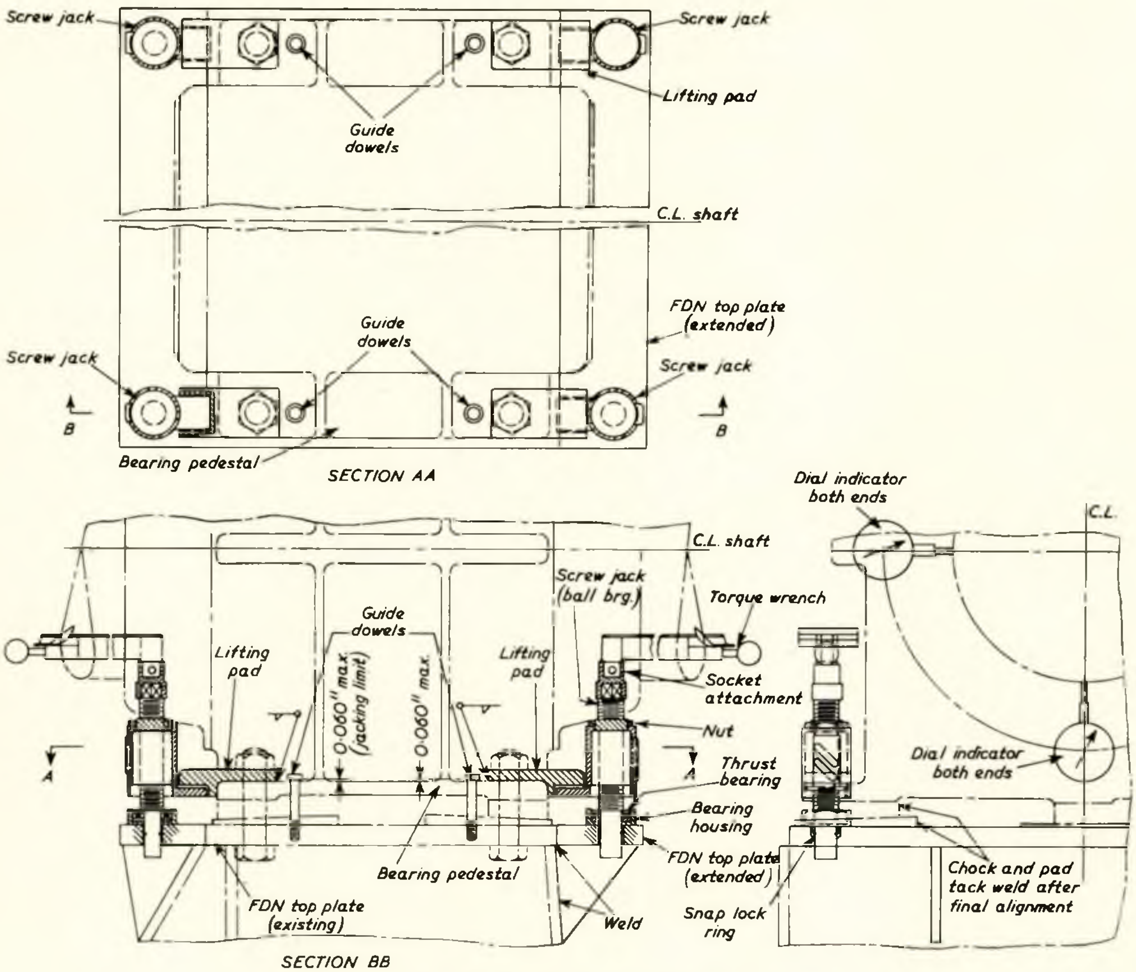


FIG. 34—*Portable load test jacks for steady bearing (mechanical)*

In the interest of simplifying and expediting alignment of propulsion equipment, he recommended that line shaft bearing pedestals be designed to allow ship's force personnel and/or repair yard personnel to adjust bearing loadings readily. To accomplish this expeditiously, the following procedure was suggested:

a) The naval architects and marine engineers would work as a co-ordinated team to establish the average loading for the bull gear bearings and all line shaft bearings in the propulsion equipment, which would ensure proper alignment. Further, this team would stipulate ranges for permissible variation of loading for each bearing (maximum and minimum).

b) Using these established loading criteria as working guides, ship's force or shipyard personnel could adjust inboard bearing loading by means of equipment delineated in Figs. 33—35 of this discussion. Figs. 33 and 34 presented methods of inserting temporary jacks between bearing pedestal and foundation, while Fig. 35 showed a permanent installation of jacks.

These jacks would be calibrated to read directly in pounds and in normal operation would not accept any of the bearing load. They would be used only to measure and adjust load. The original building or aligning activity would adjust loading by means of these jacks to the average bearing loading values determined as outlined.

Adjustments and realignment required subsequent to the initial adjustment would be accomplished by slacking off and removing holding down bolts, activating jacks, removing chocks, measuring loading as necessary, and adjusting. To simplify the adjustment procedure, the installing or aligning activity could provide ship's force with a standard set of chocks based on the average aligned condition to effect any loading

TABLE XVIII.—CO-ORDINATED ALIGNMENT DATA

Ship ....

Bearing	Loading		
	Average	Maximum	Minimum
Forward bull gear			
Aft bull gear			
No. 1 line			
No. 2 line			
No. 3 line			
No. 4 line			
No. 5 line			
Forward stern tube			
After stern tube			
Intermediate strut			
Main strut			

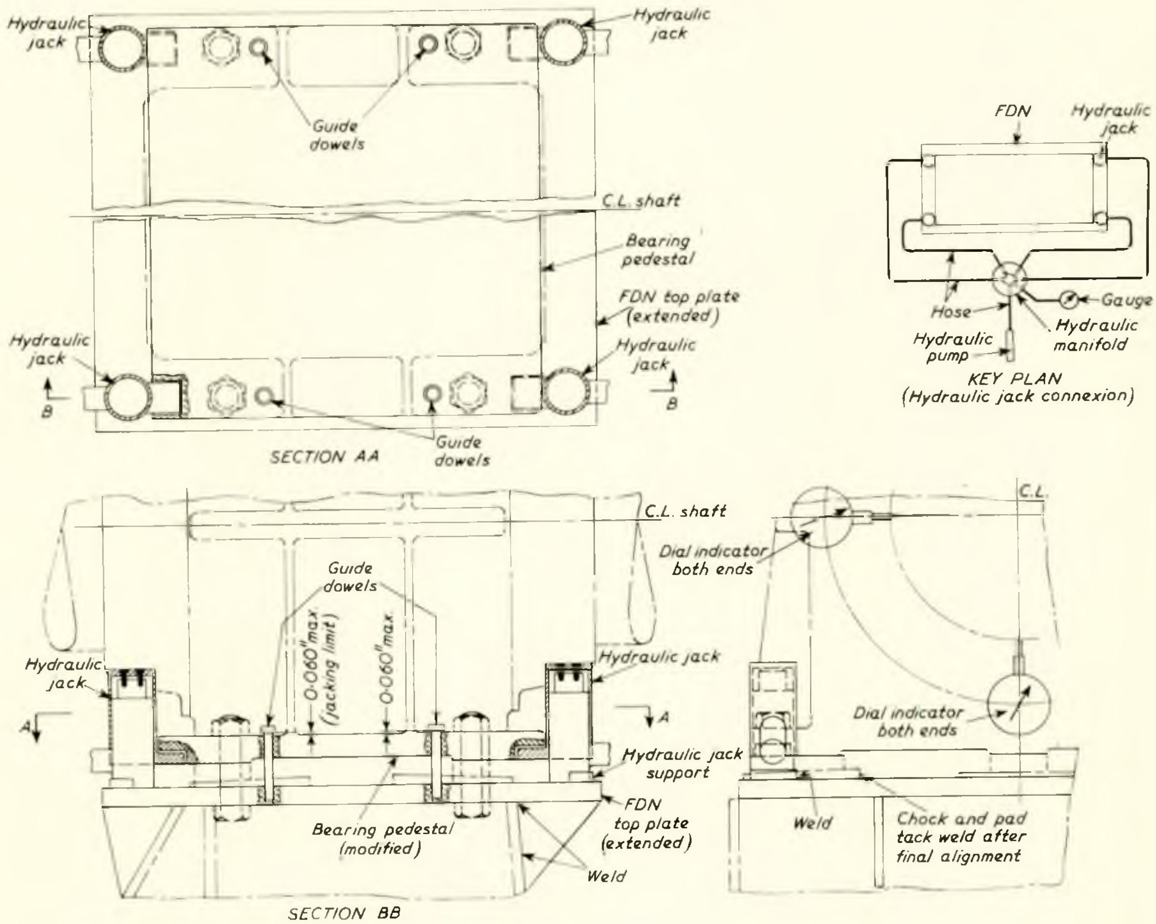


FIG. 35—*Integral load test jacks for steady bearing (permanent installation)*

within limits specified. All such chocks would be slotted and tapered for ready removal. An illustrative example of the application of this suggested system was as follows:

Table XVIII would provide adjusting personnel with the necessary control information for each bearing. Assuming that line bearing number 3 was overloaded, adjustments would be made to bring the loading within the limits of the table. Bearings numbers 2 and 4 would be checked also and necessary adjustments made to compensate for the movement made at line bearing number 3.

It was believed that the use of these readily adjustable bearings in conjunction with the improved method of gear alignment, described in the paper under discussion, could contribute to the resolution of the recurring problem of propulsion equipment alignment.

MR. T. W. DUNN (Member), MR. R. W. PEACH and MR. I. MENDELSON (Associate Members) were of the opinion that the subject of the paper was one that was belatedly receiving the attention it deserved. It would have been fitting and proper to have similar papers at this meeting, bringing out the shipbuilder's problems and those of the people responsible for maintenance and repair, as they affected the shaft alignment.

For instance, no mention was made of the abnormal loads that could occur from the Magnus effect due to lift from cross flow over a rotating cylinder. On a recent design, this load per unit length of shaft was of the same magnitude as the weight of the shaft per unit length. It should be noted that no matter where the first line shaft bearing was placed it would affect the load on and, thus, the alignment of the gear. If the initial alignment was such as to put no moment or shear on the gear flange, in the first few months of operation bearings would wear in and affect the gear bearing loads. It was true that long lengths of shafting increased the flexibility but they also increased the load on the after gear bearing.

From a practical point of view, with the effects of changing alignment due to the Magnus effect, bearing wear, hull flexure, and thermal growth, it would be well to know how much the pinions, by shifting in their journals, could accommodate gear operating misalignment.

The authors mentioned an arbitrary 0·0002in. mismatch per ft. of face width. Why should this not be, say, 0·0004in. or 0·0006in.? It might be that the pinion shifting resulted in less than the 0·0002in. per ft. in operation. If bearing wear were considered as well as hull flexure, thermal growth, bearing pressures, shaft stresses, alignment tolerances and other environmental unknowns of the ship structure, the arbitrary 0·0002in. mismatch allowed for per ft. of face width appeared to be an unrealizable goal. Furthermore, it would be better to establish the allowable difference in bearing reactions on a rational technical basis, probably as a percentage of total load, rather than the 5,000lb. to 15,000lb. in the vertical, and 1,000 to 5,000lb. in the horizontal given in the paper. Likewise, the minimum bearing reaction should be based on bearing pressures, rather than a fixed 1,000 to 2,000lb. Had the anticipated effect illustrated in Fig. 23 in fact been demonstrated as a result of shaft realignment?

Probably one of the most interesting facets not thoroughly discussed by the authors was the effect of bearing wear on alignment. They (Messrs. Dunn, Peach and Mendelson) had used the Bureau of Ships Manual, Chapter 40, for Table XIX of this discussion, although any other criteria would be equally good.

The authors' fourth case, for a twin-screw navy vessel,

TABLE XIX.—REDUCTION GEAR JOURNAL BEARING CLEARANCES

Case	Diameter, in.	Minimum, in.	Maximum, in.	Re-babbited at, in.	Minimum net drop, in.	Thermal rise, in.	Difference* in gear bearing reaction, lb.	Maximum specified load difference on gear bearings, lb.
1. Large tanker	26	0·026	0·031	0·052	0·021	0·020–0·023	48,000	14,000
2. Cargo ship (bearing 3 out)	24	0·024	0·029	0·048	0·019	0·015–0·020	15,300	5,000
3. Midship engine room	18	0·018	0·022	0·045	0·023	0·020	10,000	6,200

* Due to minimum net drop only.

TABLE XX.—WEARS AND EFFECTS ON GEAR BEARING REACTIONS

Case	Lineshaft			Stern tube			Maximum specified load difference on gear bearings, lb.
	Diameter, in.	Minimum net wear, in.	Difference in gear bearing reaction, lb.	Diameter, in.	Minimum net wear, in.	Difference in gear bearing reaction, lb.	
1. Large tanker	$24\frac{1}{2}$	0·021	63,000	$32\frac{3}{4}$	0·224	4,900	14,000
2. Cargo ship (bearing 3 out)	21	0·021	29,800	$25\frac{3}{4}$	0·223	10,300	5,000
3. Midship engine room	$17\frac{1}{4}$	0·021	11,900	$21\frac{13}{16}$	0·219	Not given	6,200

was not included, since the paper did not give the various bearing journal diameters.

It would be seen from the table that the allowable wear was of the same magnitude as the thermal rise which the authors were careful to build into the alignment.

The foregoing were the effects of the gear itself on its misalignment. From practical considerations, the shaft bearings could wear also. Table XX gave the wears and effect on the gear bearing reactions, assuming only that the forward line shaft or forward stern tube bearings wore their allowable amount.

It would be noticed that the line shaft bearing wear exceeded the allowable difference on the gear bearing loads. In the two cases given on the stern tube bearing wear, one exceeded the allowable difference and the other used up a substantial amount of the load difference.

From the foregoing discussion, it was concluded that for satisfactory application on shipboard, the details of each design must be studied and altered by mutual agreement between the gear manufacturer and the shipyard. There were some factors under the partial control of the shipbuilder; namely, loads on the bearings and bearing spacing. Likewise, the gear manu-

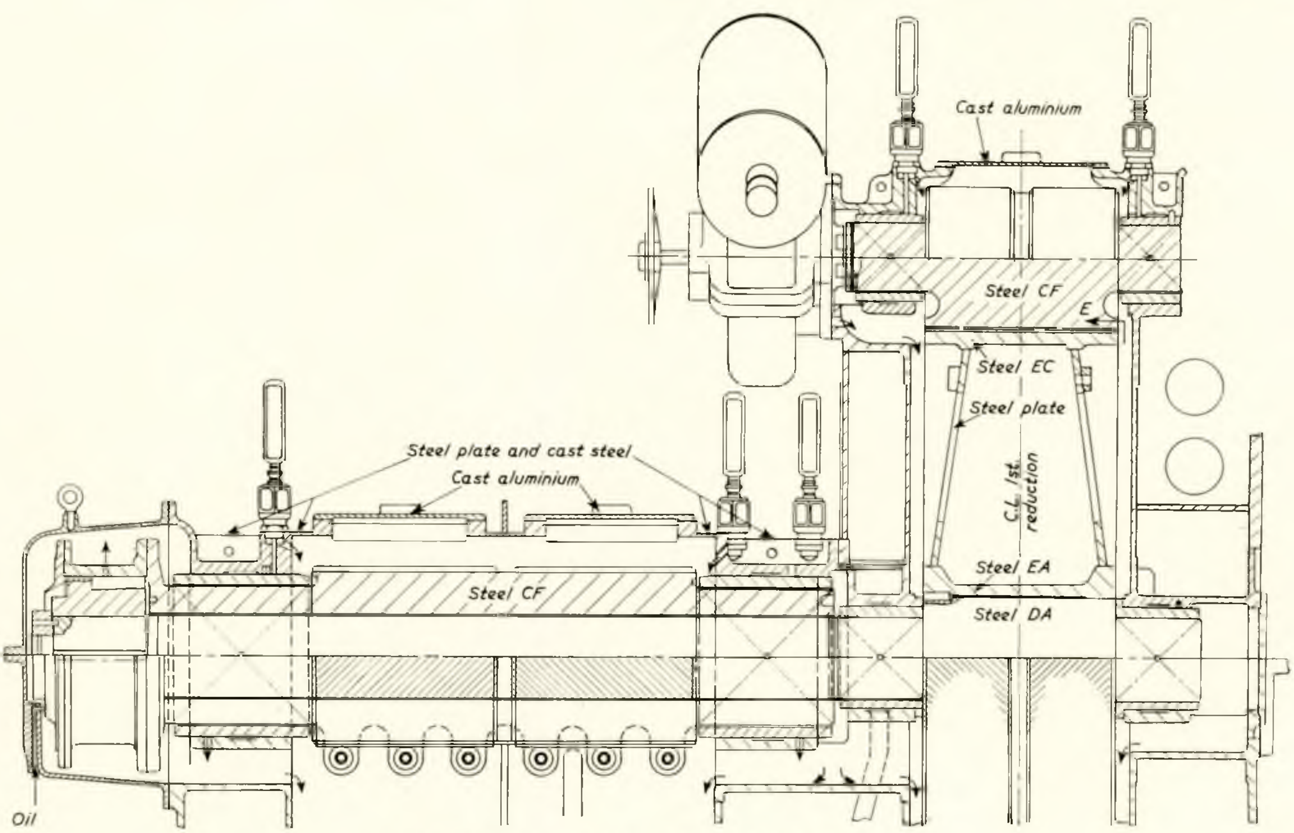


FIG. 36—*Design used in C-2 and C-3 vessels*

facturer had control of loads on the gear bearings and allowable tooth mismatch. Since the internal misalignment of pinions and bull gear was a function of their geometry just as much as it was a function of the external shafting system, it would be interesting to know whether consideration had ever been given to the other geometrical relationships between pinions and gears which would minimize the effect of unequal bearing loads. It was incumbent on both the gear manufacturer and the shipbuilder to give the customer a ship that was satisfactory in performance without abnormal maintenance, replacement or repair of bearings or gears.

MR. P. H. ENGVALL (Member) said that the problem of line shaft to bull gear alignment was indeed one to be considered carefully. Undoubtedly, poor alignment had been the cause of some second-reduction gear tooth trouble. This problem had been recognized in the past, and on many occasions the company which he represented had recommended to the shipbuilder that the distance from the aft bull gear bearing to the first line shaft bearing be increased to reduce the possibility of adversely affecting the alignment in the second-reduction elements.

The authors' approach to the problem—a rigorous mathematical analysis with the help of an IBM computer—provided the answer needed for correct cold alignment of the bull gear relative to the line shaft in terms of "sag" and "gap" to a thousandth of an inch and eliminated the hit and miss approach that had been used until now.

Several points should be kept in mind, however, when considering these values. One was the assumption that the ideal condition of all line shaft journal centres being in line and the centres of its free-hanging flanges being in line with the journals and square to this line could be achieved; and, if achieved, could be maintained. The tolerances that should

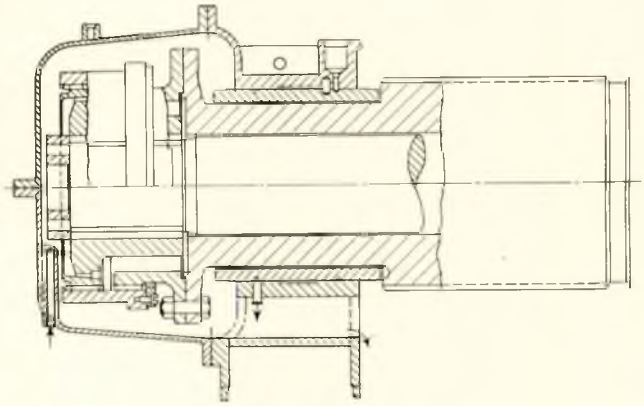


FIG. 37—*Modification of the original design*

accompany these "sag" and "gap" values due to hull distortions caused by various conditions such as cargo loading, movements of the ship in a seaway, and the "night and day" effect on expansion, and so on, could represent a good percentage of the specified offset.

The second point was the ability of the pinions to adjust their position in their own journal bearings so as to compensate for a tilt in the mating gear.

The primary purpose of this paper was to point out the serious effects that misalignment in the second-reduction element could have on the gears, and, further, how misalignment could be minimized.

He believed that it was in order to discuss another and perhaps even more serious reason for such misalignments

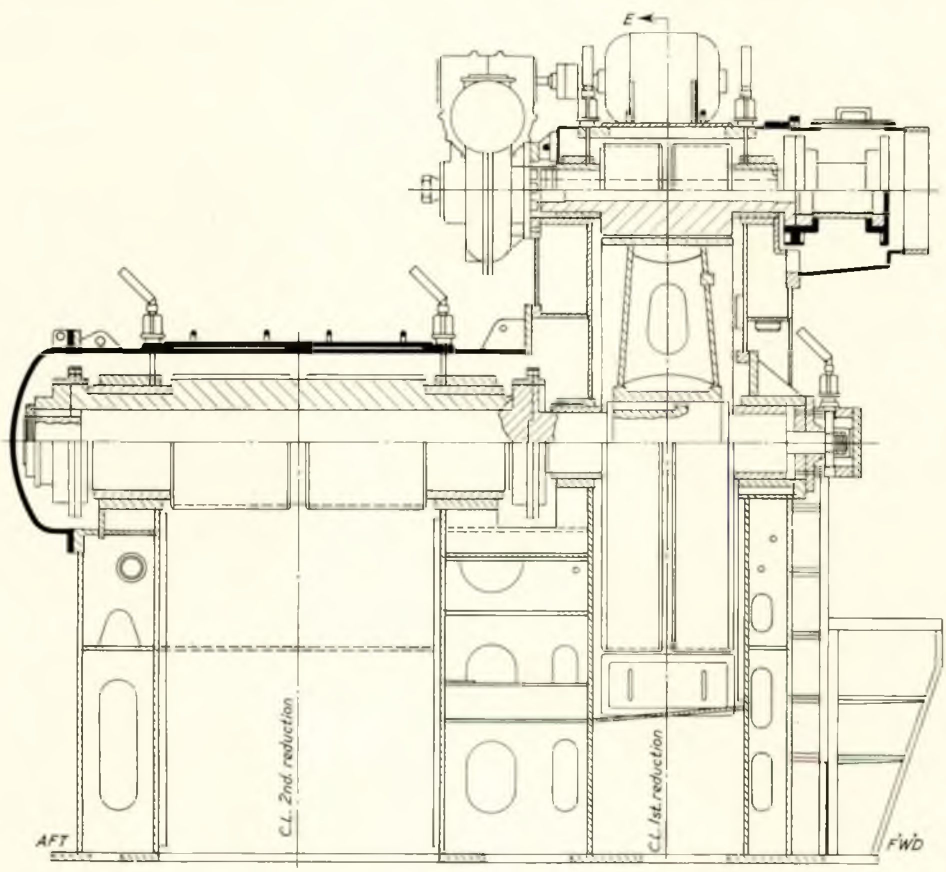


FIG. 38—*Further development of the double-engagement design*

taking place. Internal misalignment in the second reduction took place under load if full freedom of movement did not exist in the quill shaft connexions between the first-reduction gears and the second-reduction pinions. Most of the World War II cargo ships had for a quill shaft an extension of the first-reduction gear shaft passing through the second-reduction pinion. A single engaging gear tooth coupling connected the aft end of the gear shaft to the aft end of the pinion, as shown in Fig. 36. This was the type of design used in the C-2 and C-3 vessels referred to by the authors.

With this design the direction of movement of the first-reduction gear shafts and the second-reduction pinion, due to load, were not in the same direction. Therefore, a tilting moment was set up which caused the coupling to have a tendency to lock and the pinion to tilt in relation to the gear. This eliminated the freedom of movement of the gear shafts and the second-reduction pinions, and might result in galling of the coupling teeth as the coupling could not align itself properly between the quill shaft and the second-reduction pinion. It also set up resistance to the axial sliding movement that must take place in the coupling between ahead and astern operation, which, in turn, often resulted in overloading of the first-reduction gear thrust bearing.

These movements could, of course, be calculated and steps taken to reduce the bad effects of misalignment. For example, eccentric bearings could be used to obtain correct alignment under load; however, eccentric bearings had not proved to be entirely satisfactory.

Fig. 37 showed a modification of the original design, incorporating a double-engagement coupling in place of the single-engagement coupling. The double-engagement modification had been used in many gear reductions to eliminate the troubles found in installations with the single-engagement design. As far as he knew, this design change had been successful in each and every instance.

Fig. 38 showed a further development of the double-engagement design which, incidentally, was based on the same principle used in locked train gears produced for naval vessels. This design assured complete freedom of movement between the first-reduction gears and second-reduction pinions.

In conclusion, he concurred with the authors that thermal expansions should be considered when determining the alignment of shafting, gears, and turbines. He wished to point out, however, that calculated values for offset should be accompanied by fairly large tolerances, and further, that these tolerances need not be cause for alarm, since the pinions had the ability to adjust their alignment to the gear, as proved by the successful operation of modern gear installations.

Mr. B. B. Cook, Jr. (Associate Member) said that the subject of proper alignment of shafting, gears and turbines, of course, was not new. In fact, the need for a standardized guide to reduction gear and shafting alignment had been realized by the Society for quite some time; and Panel M-12, of which one of the authors and he himself were members, was in the process of preparing such a guide.

It was believed that the most important contribution of the proposed method of analysis and alignment of the line shafting to the propulsion gear might be that it would act as a stimulus for further thought.

The writer was of the opinion that the authors had over-emphasized the importance of gear tooth trouble resulting from misalignment in the second-reduction gear mesh due to improper line shaft design or alignment.

The desirability of tending to equalize the static loading on the bull gear bearings, and particularly maintaining a positive downward static loading on the forward bearing, had been known for several years, and had resulted in many discussions with shipyards by the writer's company in order to arrive at a mutually satisfactory shafting to gear alignment.

The authors had attempted to set a rather stringent limitation on the amount of misalignment that was permissible in the second-reduction gear mesh. It was undoubtedly desirable to arrive ultimately at a definite limitation; however, the writer believed that additional information and other considerations were required before such a limitation could be set with any degree of exactness. While this study was undoubtedly correct for the theoretical condition, one questioned its practicality, based on the following considerations:

a) It was believed that the authors had not considered that when a bull gear was tilted with respect to its bearing bores, the out-of-plane condition of the axis of the gear and one pinion, and the unequal centre distance from the forward to aft end of the other pinion tended to concentrate the load on the end of the helix, which in turn caused a larger bearing reaction on that end of the pinion, and the pinion axis tilted in a direction to compensate partially for the tilt in the bull gear.
b) It should also be realized that it was impossible to have all the bearings located on a true straight line; even if it were possible to accomplish this, the bearings would not retain this condition due to variation in temperature, loading and flexing of the hull.
c) The authors had considered parallel thermal expansion of the foundation; and this, of course, was not necessarily true. In fact, the main reduction gear foundation might expand more at one end than the other, depending particularly upon the type of foundation structure and the location of the lubricating oil sump. It was believed that this was the area in which more detailed test information was required before a finite value as to misalignment could be reasonably established.
d) Closely associated with the preceding comment was the need for further test information pertaining to the thermal growth of the line shaft bearings compared with the growth of the main reduction gear foundation.

With regard to the thermal expansions of turbines and gears, he was in agreement that a good alignment was necessary, and was an important factor, contributing to longer coupling life and eliminating a possible cause of turbine vibration.

In conclusion, it was the opinion of the writer that the authors were to be congratulated for their comprehensive paper, but it should be accepted with reservations, particularly with respect to the line shaft-to-gear alignment; since it was believed that the conditions had been idealized to such an extent that in practice a good alignment could be obtained without necessarily complying with the proposed limitations.

Mr. P. E. Atkinson (Member) said the problems involving alignment of main reduction gears and related line shafting and turbines warranted a great deal more attention than they had received in the past. Perhaps the engine builder and the shipbuilder had not taken full advantage of the other's knowledge and experience with respect to this matter. The authors had approached the problem from fundamentals and should be complimented on the care and thought which had gone into the preparation of this fine paper. More important, however, the stimulation of thought and discussion that the paper could not help but arouse could only result in healthier and sounder main propulsion systems in the future.

At Sun Ship, they had carried out some preliminary experiments on recent tankers in an effort to determine the bull gear bearing static loadings and the position of the bull gear shaft under full load conditions. Although the bearing loadings could be reasonably well determined statically in the "hot" condition, they did not believe that this information necessarily would reasonably indicate the position of the bull gear shaft under operating conditions. In fact, preliminary experimental data seemed to indicate that other factors than those considered in the paper substantially affected this shaft position and thus the tooth contact. More precise instrumentation could and was being applied to obtain the required information more accurately. Consideration was being given to installing instrumentation permanently on one vessel to determine the effect of stern bearing wear-down on gear alignment.

There was a distinct danger after reading this paper of reaching the conclusion that the alignment problem could be resolved easily by following the proposed method. Unfortunately, this was an over-simplification of the problem. Although the authors pointed the way to the solution, several factors which might have an important influence on alignment had not been resolved satisfactorily. Anyone who had checked shaft alignment in the cool morning and again in mid-afternoon with the hot sun beating on deck, could not help but wonder at recommended shaft alignment changes which involved only a few mils. Actual data on the rise of the bull gear journals from cold to hot operation condition with various types of main engine foundations were required to verify the present broad assumptions which varied quite radically among the engine manufacturers. Data under operating conditions were required on the actual position assumed by the bull gear shaft in its bearings before the indicated calculations could be accepted as reliable.

Generally it had been the experience of the writer that a ship's structure deflected a surprising and somewhat unpredictable amount in a seaway under full power conditions. Furthermore, extreme (in terms of mils) structural deflexion occurred owing to varying ballast and temperature conditions encountered in everyday operation. Precision machinery under these conditions deserved every chance it could get to assure proper alignment, but they could not afford the luxury of assuming that they were operating on a level stationary non-deflecting base, since such a circumstance did not exist in the dimensional realm under consideration. In view of the foregoing, it would seem to be inevitable that marine gears would encounter some degree of misalignment and must be designed accordingly. Efforts to comprehend more fully and accurately the factors contributing to misalignment must be continued and a closer understanding of mutual problems between the engine builder and the shipbuilder must be encouraged.

Mr. E. B. Williams (Member) felt he was far from being qualified to enter into a technical discussion of this valuable paper. However, a small group of their Machinery Department did give this paper some study and his remarks were mostly the result of their efforts.

The observations and conclusions contained in the paper illustrated how successfully engineering logic, fortified by adequate calculations, could provide answers for such puzzling discrepancies as the difference in gear tooth alignment between that existing in factory assemblies and shipboard installations, where apparently every effort had been made to have duplicate conditions. One could readily appreciate that shipboard shaft alignment might not be as precise an art as alignment of gears within a case when one reviewed the magnitude of movement of journals in a bearing responsible for unsatisfactory tooth contact.

The shipbuilder working in an atmosphere provided by Mother Nature on a constantly changing support, far removed from the concrete base supporting a factory assembly bed, and using tools which must be transported from job to job, must be content with a degree of perfection falling far short of that attained by the gear manufacturer. However, in spite of these hazards, the old methods of alignment commonly used in the shipyards resulted in tooth contact which was frequently quite good. Last year his company aligned the short shafting of a Lake ore carrier in the established manner, and, later, bearing loads were measured by means of calibrated hydraulic jacks by associates of the authors of this paper. Calculations of the type referred to had been made previously. The total bull gear bearing load was about 38,000lb. There was some surprise when it was found that individual gear bearing loads were within 4,000lb. of each other, with the heavier load on the forward bearing, although line shaft bearing loads were quite different from the calculated loads for initial conditions.

The shipbuilder often found steady bearings and stern tube bearings to be loaded contrary to design calculations, particularly as shown by bearing wear after several years of operation. In most cases no real harm ensued. In addition to the changes in alignment as a result of wear of the usual stern tube bearing materials, shaft bearing loads were affected by the varying loads on the hull structure, uncontrolled temperature changes in the steel structure, and the uneven thrust loading of the propeller during a revolution and for different draughts. The shafting system tolerated these changes. A bearing pedestal support might easily vary in temperature from 50 to 80 deg. F. in an operating season on the Lakes, changing the vertical position of the shaft by 0·015in. On the other hand, the temperature of the gear case was properly kept within close limits by regulating the incoming oil temperature, so that wall temperatures given in Fig. 30 of the paper might be expected throughout the operating season. The effect of the uncontrolled temperature variation in the bearing supporting structure alone was beyond the proposed limits.

The need for more understanding of the effects of varying bearing loads and different shaft alignments was apparent because of the possible results. The apparent offer made by the authors to supply complete calculations as a guide to alignment, was welcome, but to translate this information into requirements that the shipbuilder must meet, might be premature. Shipbuilders might be expected to be wary of additional costs and delays affecting shafting and gear installations. Weights of propellers, shafting and couplings were often estimated for calculation purposes and might change frequently along with line shaft bearing locations during design stages. Although there was fairly good agreement in calculations made for the in-line cold condition, there was rarely any real agreement for other conditions. Calculation discrepancies were the result of different assumptions made by the individual calculator. So far as they knew there was no published standardized method for calculating bearing influence factors. More information concerning the derivation of these factors would be appreciated, particularly because in most cases measurement by hydraulic jacks of the loads on both bull gear bearings or the stern tube bearings could not be made directly. Furthermore, jack readings for various shaft elevations would be expected to agree with previously calculated bearing influence factors and this agreement might be difficult to demonstrate.

The greatest gain in better shipboard shaft alignment appeared to be in the improvement of gear tooth contact, by maintaining the gear in its proper location in its oil wedge-type sleeve bearing. This type bearing might be inconsistent with the degree of precision now attained in gear manufacture. Should this bearing be changed to the self-aligning roller bearing type? They understood that such bearings positioned the shaft from 0·00090 to 0·00140in. The additional cost of such bearings was admittedly high but possibly much less than corrective action for poor tooth contact or compared with the shipbuilders' cost in seeking a finer shaft alignment than permitted by factors beyond his control.

The paper contained much valuable information and its worth would increase as it was absorbed by shipyard design and installation personnel. It was a challenge to the shipbuilder to strive for a higher degree of perfection so that his contribution would complement the excellence of the gear unit he installed.

Mr. J. H. Lancaster (Member) said the subject of shafting, gear, and turbine alignment was truly one of diverse opinion and experience. The clarification and standardization of procedure which might eventually result from such thought provoking papers as this would be of great help to all concerned. It would seem reasonable, however, to assume that general acceptance of such a standard procedure and its criteria should result only after critical examination of its basic assumptions, conclusions, and correlation with industry-wide service experience.

The service experience of the Bethlehem Steel Company over the past 10 years differed from that of the authors concerning incidence of wear and pitting on second-reduction gear elements. Records indicated that no wear or pitting had occurred on any merchant second-reduction gear element powered by company turbines. This period included the production of 120 merchant units ranging from 7,700 to 27,500 s.h.p. with a total output of 1,715,450 s.h.p. Of these, 18 7,700-s.h.p. and four 15,000-s.h.p. units were of the articulated type, the remaining 98 units being of the nested type. Ten sets were installed in the Mariners and four sets on the *Independence* and *Constitution*, the remaining being tanker units.

The Bethlehem computer method of calculation of shaft alignment and thermal growth assumptions were basically the same as those used by the authors. This was also true of the method of actual alignment wherein gaps and sags were used. However, their experience to date was that calculated ratios of aft to forward main gear bearing reactions in the order of 2 to 1 in the hot condition had not produced any abnormal wear pattern or resulted in any distress. Hot condition differences in main gear bearing reactions in excess of 50,000lb. had caused no difficulty. Negative main gear bearing reactions in the cold state had caused no trouble.

The foregoing was not cited as a recommendation to create deliberately large differences in main gear bearing reactions. Their current practice over the past years was to optimize bearing locations and reactions with the aid of their 650 computer. However, it did appear that the authors' criteria were unduly pessimistic in favour of the gear and its bearings. For the hot condition, it would appear that a ratio of main gear bearing loads of 2 to 1 should be acceptable and that negative loads in the cold state should be permitted. It was also recommended that calculated gear bearing pressures be allowed to increase to 200lb./sq. in. if line shaft reactions in the hot state were added thereto.

The foregoing might appear too permissive when gauged by the crossed-axes conditions developed in the paper. It should be recognized, however, that some of the authors' assumptions might not be strictly in accord with actual conditions. First, precise knowledge of thermal growth was assumed. It should be noted that a 10 deg. F. temperature differential in a length of 4ft. would result in a 0·003-in. length differential. Fig. 30 indicated that such differential temperatures did exist. This was more than the main gear out-of-plane movement cited by the authors for a 30,000-lb. difference in main gear bearing loading for a large tanker. Also, there was no assurance that this calculated hot condition was one of reality or permanence. In fact, the probability was that it was not. When a ship was waterborne at a shipyard, it was normally in a light condition when shaft and gear alignment was made. The combinations of water, ambient, and upper deck temperatures might differ considerably from those to be encountered in service. The condition of loading was certainly not the same. These effects, not to mention the dynamics of wave action, were measurable in inches of hull deflexion rather than in thousandths. The vibratory double-amplitudes from propeller-excited hull and longitudinal criticals could be in the order of 0·050in. and more. Furthermore, it should be noted that the development of the crossed-axes geometry was based on an equal division of load between forward and after pinion helices. It would seem more realistic to assign a permissible variation in loading to each helix where the designed K-factors were not marginal. This would substantially reduce the calculated amount of plane mismatch.

It was rather surprising that the record of the articulated gears cited by the authors should indicate a relatively greater sensitivity to variation in shaft alignment in comparison with nested gears installed by the writer's company. In the past 10 years, considerable mention in technical and other publications both there and abroad had been made of the theoretically superior ability of articulated gears to absorb and compensate for misalignment. There might be a possibility that the difference in thermal expansion between the articulated gears' aft and middle walls and supporting ship structure carrying the second-reduction pinions might have contributed to the difficulties recorded by the authors. In a nested-type gear no middle wall existed, all support walls being external and subject to symmetrical thermal expansions. Comment by the authors on this point would be appreciated.

To summarize, it was believed that the criteria for shafting-to-gear alignment used by the authors were excessively stringent. Although the mathematical relationships established in the paper appeared to be quite valid, it was believed that factors such as static, thermal, and dynamic hull bending, vibratory displacements, and unequal gear and foundation thermal expansions could be of greater significance than the shafting weight and moment effect. It was also believed that, for gears operating at conventional design K-factors, a tolerance existed for division of load between forward and after-pinion helices in excess of tooth-to-tooth spacing allowable error. It therefore seemed reasonable not to depart from a broader main gear-bearing loading tolerance substantiated by successful experience to a limitation based on theoretical parameters of gear sensitivity alone. It was recommended that an investigation of the effect of other variables affecting gear loading should also be undertaken before any highly restrictive standard was adopted.

MR. J. C. REID, JR., considered that the paper very capably presented a topic worthy of careful consideration. It should help solve some of the problems connected with the design, installation and maintenance of ships' propulsion machinery. As the title of the paper indicated, all of the elements that made up the entire system were treated as a co-ordinated, continuous unit. This was one of the few papers on the subject of alignment that treated the entire shaft system as a continuous elastic member. Too often, short-cut solutions were made which lacked this co-ordinated approach. The importance of gear alignment was emphasized but not so as to overshadow the need for co-ordinated consideration of all other elements of the system. The authors were to be complimented for this very excellent paper.

He desired to comment on specific portions of the paper as follows:

a) The paper discussed the substitution of an oil lubricated metal line shaft bearing for a water lubricated, forward stern tube bearing on the port shaft of a navy twin-screw ship. Such a change had been made on several recent classes of navy ships. These changes were made after it was determined that the vertical position of the forward stern tube bearing had a very sensitive effect on the reaction of adjacent bearings, hence an adjustable bearing design was sought. It was obvious that the vertical position of a line shaft bearing could be adjusted without drydocking the ship. This was not the case with a stern tube bearing. Forward stern tube bearings were not eliminated because of dissatisfaction with their wear characteristics.

b) He submitted further comment on this example of the navy ship by citing that on three classes of such ships the system was changed from a seven-bearing to a six-bearing arrangement. That was, the forward line shaft bearing (number 3) was eliminated in entirety because it created a system too sensitive to vertical position changes of this and adjacent bearings. It was presumed that the ship exemplified represented a fourth class, whose details he had not personally reviewed, wherein this seven-bearing arrangement was determined to be relatively limber and less sensitive to bearing vertical position changes.

c) The paper suggested that wick or ring oiled line shaft bearing pressures should not exceed approximately 40lb./sq. in. It was conceded that this was a valid limitation for wick oiled line shaft bearings. In fact it was preferred that wick oiled bearings should not be used on navy ships. As regards ring

TABLE XXI.—EXAMPLE

L/D ratio	P, lb./sq. in.	N, r.p.m.	h, in.	f	N, r.p.m.	h, in.	f
1	80	120	0·002		240	0·004+	
1½	53	120	0·003+	0·002	240	0·006−	0·002+
2	40	120	0·005−		240	0·008	

L = bearing length, in.
D = bearing diameter, in.
P = bearing loading, lb./sq. in. (on projected area).
N = shaft speed, r.p.m.
h = oil film thickness, in.
f = friction coefficient.

oiled line shaft bearings, it was contended that a 40lb./sq. in. loading limitation was conservative for the ordinary application. This contention bore discussion. A hypothetical case had been set up for bearing number 3 of the large tanker exemplified in Fig. 14 of the paper. Taking the condition giving the 49,250-lb. reaction in column 6 of Table V, they assumed typical viscosity, temperature, clearance and other bearing relationships and calculated characteristics as shown in Table XXI.

It was contended that approximately 50lb./sq. in. is a conservative design figure for ring oiled line shaft bearings. He was inclined to be more concerned with a high ambient space temperature than with high bearing loadings. The foregoing hypothetical problem assumed a shaft alley ambient air temperature of 80 deg. F. in which good bearing heat dissipation assured. There were certain extraordinary applications, such as where line shaft bearings were located in fire rooms with high ambient temperatures, where a second look might be required. However, good fortune was frequently with them in that fire room located bearings, designed to be the same size and interchangeable with all other line shaft bearings, were frequently lightly loaded. When this was not so, forced air blowers, or other means of artificial cooling might be necessary.

The authors were to be congratulated for so capably presenting this timely topic. Much benefit would accrue by its dissemination to engineers engaged in the design, installation and maintenance of ships' propulsion machinery.

MR. C. L. LONG (Associate Member) said many factors might influence the internal alignment of a reduction gear and these were:

a) The state of completion of the welding on the ship at the time the gear alignment was made.
b) Whether the alignment was made with the ship on the ways or in the water.
c) Temperature distribution of the hull at the time the final alignment was made.
d) Racking of the gear case and rigid body movements of the gear relative to the line shaft bearings resulting from the operating torque, thrust loads and sea and cargo loadings.
e) The thermal expansion of the slow speed gear bearings relative to the line shaft bearings and the thermal expansion of the gear bearings relative to each other. It should be remembered that the centre wall of the reduction gear would expand more than the outer walls.
f) Redistribution of tooth load due to centrifugal and temperature effects acting directly on the pinion and gears. It had been postulated by one gear manufacturer from carefully conducted tests that, depending on whether the pinion was driving up the vee formed by the two helices or down the vee, the centre of the pinion would take more or less load. If the driving direction were such that the centre of the pinion took more load, then a regenerative heating effect took place resulting in extremely heavy contact in the centre of the mesh.
g) The hydrodynamic properties of gear journal bearings in aiding or hindering the ability of a gear to adjust itself to various running conditions.

Most of the foregoing factors and considerations did not lend themselves to precise analytical solutions. Reduction gear installation and alignment were still more of an art than a science. This did not mean that the problem of gear alignment should not be investigated analytically but that experience was an all-important asset.

In this paper the oil film thickness and running position of the journal in the bearing had been neglected. It was felt that these two factors might have some significance and should be considered. The authors' comments would be appreciated as to what effect their studies had shown the hydrodynamic properties of the bearings to have on the alignment procedures outlined in the paper.

Thermal rise of the reduction gear and gear foundation was the major contributing factor causing the unequal loads on the slow speed gear shaft. Their experience and checks would indicate that the rise was less than indicated in the paper.

Temperature readings were taken on a large tanker reduction gear foundation and lower gear case. These temperature readings showed that a considerable difference in temperature existed between the inner and outer walls of the lower gear case and between the top and bottom (bottom being hotter) members in the foundation adjacent to the slow speed gear wheel. Members in the foundation not bounding a space containing the reduction gear oil were more uniform in temperature between top and bottom and lower in temperature. A mean effective temperature of 102 deg. F. and 105 deg. F. was estimated for the foundation and lower gear case, respectively. The thermal rise was estimated to be 0·016in. This gear and foundation were probably similar to the large tanker discussed in the paper, where the total thermal growth was taken as 0·020in. to 0·025in.

In a similar high powered tanker, calculations of the bearing reactions and influence members showed that the reaction of the first steady bearing would go to zero if the slow speed gear rose 0·018in. A check was made of the bearing reaction by rolling the lower half of the bearing shell out and measuring the sag of the shaft at the bearing with the gear both cold and hot. Although this method was not considered extremely accurate, the results showed that the gear rose much less than 0·018in.

It was felt that the only way to pin down the actual thermal rise of the slow speed gear was to measure the slow speed gear bearing and first line shaft bearing reactions with the gear cold and hot.

A comment in regard to the authors' remarks concerning the negative reaction shown in Table V for bearing number 7: these reactions were static reactions and did not take into account the effect of propeller thrust. For single-screw ships, with the propeller completely submerged, the resultant propeller thrust was located in the first quadrant (looking forward) at full power. This thrust moment tended to offset the gravity moment of the propeller, which increased the reaction on the forward stern tube bearing. Also, they had found with the hard stern tube materials, lignum vitæ, and the like, that when the propeller was installed on the propeller shaft in drydock the propeller shaft did not rise in the forward stern tube bearing, indicating that the resultant after bearing reaction was well aft in the bearing.

Mr. H. W. Semar (Member) said that in post-war naval ships and high power merchant ships, it had become common practice to locate the main propeller thrust bearing immediately aft of the main reduction gear. If alignment procedures were used which ignored the deflexion of the thrust shaft overhung from the gear, there would result a much more severe unloading of the forward bull gear bearing than would occur if the gear coupling were close to the gear bearing.

For this reason his company had adopted alignment methods similar to those described even though they had not observed any gear difficulty which could be traced to this source. They could see, however, that neglect of this factor could lead to trouble in any case because of the effect of oil sump temperature in raising the two gear bearings. They were therefore in general agreement that the procedures outlined for the slow speed shafting were proper and should be given consideration early in the design stage before bearing sizes and locations were fixed.

Commenting on the conclusions reached:

The 5,000 to 15,000-lb. differential for vertical components of gear reactions was reasonable. On large gears the upper value should be satisfactory. The 1,000 to 2,000-lb. horizontal figure appeared unnecessarily small.

The 150lb./sq. in. maximum for static loading of large gear bearings appeared high. They would suggest 100lb./sq. in.

The stress in the gear shaft should be such that it had a greater factor of safety than the adjacent line shaft rather than a prescribed maximum.

The permissible mismatch of 0·0002in. per ft. of face width was believed to be somewhat low.

The paper further described the turbine-to-pinion alignment problem. They concurred in the methods for estimating the setting of the turbines to their pinions to minimize the duty on the high speed couplings. They pointed out, however, that structure temperatures could not be predicted accurately. In the example illustrated, an error of only 10 deg. F. in estimating structure temperatures would result in an error of 0·009in. in the l.p. turbine position because of the great distance (over 11ft.) through which differential expansions took place. It would be preferable to mount the l.p. turbine to the gear case at a higher elevation so that the effect of temperature differences could be reduced by as much as two-thirds.

Mr. C. C. Atkinson (Member) said the authors were to be congratulated for shedding light on heretofore somewhat mysterious and unexplained tooth distress frequently noted on second-reduction meshes.

The writer could confirm the authors' observation that gears on C-2 and C-3 ships had shown lack of uniformity of contact on low speed elements. This condition had appeared on the gears made by manufacturers other than that of the authors as well.

The saving feature on these older gears no doubt was the relatively soft materials used. While pitting did occur, breakage was relatively rare. The wearing characteristics of the softer material resulted in more uniform contact after a period of service even though no steps were taken to improve the poor alignment. Also, the occurrence of pitting served as a warning to make alignment adjustments. Furthermore, the conservative design permitted the gears to perform satisfactorily even with concentrated loading. Extensive load testing of World War II-type gears at the U.S. Naval Boiler and Turbine Laboratory had demonstrated the capacity of these units to carry up to three times the designed load.

The writer desired to inject a note of warning that improper alignment between the bull gear and the line shafting was potentially more serious with modern type gearing than on World War II units. The contention was based on the use of harder steels and finishing methods which minimized wear and pitting; thus, occurrence of crossed-axes contact was not readily observed by visual inspection until after considerable time of operation had occurred. While many advantages accrued from the use of advanced methods of gear manufacture, the greater premium was placed upon proper initial bull gear-to-line shafting alignment. Thus, the importance of close co-operation between gear manufacturer and shipbuilder, as propounded by the authors, could not be over-emphasized.

The solution of the problem appeared to be in achieving an alignment such that excessive difference in bull gear shaft bearing loading did not occur. This suggested to the writer that a convenient method of measuring low speed bearing loading might well be employed. Provision of suitable instruments for use at least during the trials for measuring bearing loading at or near full power operation was considered feasible and highly desirable. Suggested means were: (1) use of strain gauge load cells incorporated in low speed bearing shells; and (2) strain gauges mounted on line shaft sections in conjunction with slip rings to measure bending moments. Both methods had been employed by the U.S. Naval Boiler and Turbine Laboratory in connexion with full scale machinery tests to study the effects of excessive misalignment in the line shafting. It was pointed out that these studies at NBTL were primarily intended to obtain noise data. Test data obtained, however, did have important secondary application as regards bearing moments and crossed-axes contact of second-reduction gear elements.

The authors' treatise on alignment of turbines to gears was quite pertinent to minimizing wear of dental-type flexible couplings. Minimum misalignment at normal operating conditions was all-important to maximum coupling life. Little doubt could exist that good coupling alignment was indicated for minimum vibration also.

Captain Ivan Monk, U.S.N. (Member) said that some time previously the chairman of their Machinery Committee appointed a panel to investigate the internal alignment of ships' reduction gears. Their ultimate objective was to develop a guide, containing recommended procedures for reduction gear alignment.

During their Panel meetings each member described the gear alignment procedures of his particular gear company or shipyard. As might be suspected, each was different from all others. However, all of them had one thing in common: all were based upon the premise that the gears were once properly assembled and accurately aligned in the manufacturer's plant. The various measurements and readings made at the time of reassembly in the shipyard were expected to duplicate the original alignment conditions in the manufacturer's plant. These expectations did not always materialize.

In reducing the various gear alignment procedures to a common denominator, they prevailed upon Mr. Andersen, one of their Panel members, to undertake the job of editing the Guide.

A few weeks ago he had called Mr. Andersen on the telephone and asked him how he was getting on with the Guide to Reduction Gear Alignment. He said: "Fine. I am working on the most important part; I am working on the line shaft alignment".

In this fine paper the authors had shown how inseparable and how interdependent were the internal gear alignment and the external line shaft and turbine alignment. Their paper was an invaluable contribution to their knowledge of this complex and controversial subject.

Discussion Held at The Institute of Marine Engineers, London, on 26th November 1959

Mr. T. W. Bunyan, B.Sc. (Member of Council) said he was honoured at being selected to open the discussion on what he considered to be a most important paper.

He thought it was high time that the Institute had a paper in which the somewhat vexed problem of bearing and shafting alignment was critically dealt with. The paper most adequately highlighted some of the hazards which might arise from what might appear on the surface to be satisfactory alignment. The fact that the authors had had to use a computer for the purpose was good common sense, and he would like to know that the programmes had in fact been released to the I.B.M. Company, in which case similar calculations might be made very quickly and cheaply.

He admitted he could not suppress a smile when he first read the paragraph on page 149 in which the authors catalogued the data they furnished to the shipbuilders, which laid down, amongst other things, the permissible bearing loads in the vertical and the horizontal planes and the maximum permissible bending stresses in the bull gear shaft, but on rereading the paragraph, he found that it made good sound sense. The requirements had, in fact, specified just those prerequisites that would ensure good alignment.

He could not agree more with the authors that a measurement of the actual bearing loads in the hot and cold condition, with the shafting coupled together, was probably the best indication that the calculated alignment had in fact been achieved. Whether this was practicable or not was another matter. As the authors made quite a point of this matter in their paper, he would be grateful if they would define exactly how bending loads were measured in practice.

From his wide experience of gearing failures in a variety of ship types, he had very seldom come to the conclusion that alignment had, in fact, been the cause of trouble. Materials, design, axial and torsional vibration, and gear cutting inaccuracies were the most common causes. He did not say this to detract in any way from the value of the paper. He could hardly dare ask the authors to give of their own experiences, because, as manufacturers who had for years enjoyed a very fine reputation for gearing design and high precision gear cutting, he knew beforehand what their answer would be, but he could ask them whether gearing trouble caused by misalignment was in fact a significant issue in their opinion.

He wanted to ask, as the authors did not deal with the matter in their paper in any great detail, that in view of the very considerable flexural changes in the ship's hull from the loaded to the light condition, which might amount to anything up to an inch in a one-hundred foot run, did the authors specify any loading condition before an alignment check was made? Such a loading condition might be deduced as a result of measurements they had themselves made under varying loading conditions and in various types of ships. For instance, the variation in alignment of a large Diesel engine bedplate might be of the order of 180 thousandths of an inch from the loaded to the light condition in a midship engined cargo ship. The deflexion however was usually a smooth continuous curve, and a compromise could be made in the relative bearing heights, depending on the loading condition at the time of measurement.

The fact that his company had avoided geared drive in the large passenger ship now being built by them, should not be construed as a vote of no confidence in gearing. There were other important considerations which had won the day.

Concluding his remarks, he thanked the authors for an excellent paper, which he said members would be proud to include in their journal.

Mr. S. Archer, M.Sc. (Member) said that marine gear designers, in common with those engineers whose job it was to install and operate high power propulsion gearing, must welcome this fine study of the alignment problem in geared turbine installations for a number of excellent reasons. Firstly, because it was doubtful if anything on the subject anywhere near approaching this paper in scope and extent of detailed numerical calculation had ever before been attempted; certainly not in this country. Secondly, because the examples chosen were eminently practical and representative of current merchant and naval propulsion systems. And, thirdly, because the authors brought out very clearly in actual figures a disturbing effect which was common to all rigidly coupled double reduction gears, namely, the misaligning action of the bending moments imposed upon the main wheel shaft bearing by the thrust and line shafting.

The paper was also an excellent example of the way in which by modern computer methods problems could be solved which formerly might not have been attempted because of the limits of time and patience which had to be set, not to mention cost in man-hours.

The methods of influence numbers was both a powerful and practical way of tackling the variation in bearing load distribution caused by changes in alignment, and he was quite certain that if similar methods could readily be made available to designers throughout the world through this computer technique, they would be very grateful indeed.

The first example of a large tanker installation brought out a number of points. The large negative downward reaction at the forward end of the stern tube was characteristic of this type of shaft set-up and might well contribute to the tendency to lateral whirling type vibration, which had come so much to the fore in recent years.

He had tried lowering the stern tube bearing 30 thousandths, as the authors suggested, and he had found that it did, in fact, almost wipe out the negative bearing reaction at the forward end of the stern tube, coupled with a reduction in the stern bush loading itself. This, to his mind, was a demonstration of the possible gains to be derived from slope-boring of stern tube bearings. He wondered whether anybody in this country was trying it, as it certainly seemed to have its points, and whether the authors were for or against such a procedure. He said it would be interesting to hear their views on it. Incidentally, could the authors state what condition of loading of the ship was recommended for checking the initial alignment of gears and shafting?

In most single-screw tankers the arrangement of the second-reduction gears was as shown in Figs. 12 and 13; that is to say, with the high pressure pinion to starboard, whereby the authors had shown that it was the low pressure pinion which seemed to suffer most every time. That was particularly true where the thrust block was abaft the gears, as it tended to throw the balance between the two main wheel shaft bearing reactions badly out of equality.

He said he noticed that in Fig. 11 of the paper the forward end of the helix would be open 0·65/1,000in. for a 30,000lb. heavier load on the after main wheel bearing, whereas the high pressure alignment would be satisfactory.

It would appear that when the pinion body torsional and bending deflexions were taken into account—which for some reason the authors seemed to have overlooked—the extra tooth opening at the forward end of the after helix (assuming that the quill was driving from the aft end) might be anything from 0·60/1,000in. to 0·001in. and in a sense to aggravate the condition due to main wheel skew. It was, therefore, not surprising that of the many gear sets in large tankers that had fallen to his lot to examine in recent years, he had found that the low pressure pinion bedding always showed hard bearing, very often at the aft end, i.e. the driving end of the pinion, and so he was inclined to think that was perhaps some further evidence tending to confirm the truth of the authors' contentions. He asked if the authors could please state the face width of the gears in the tanker installation in order to permit the assessment of pinion body deflexions.

Referring to the footnote to Table I, could they take it that the ϕ on the left-hand side of the equation for pressure angle should in fact be ϕ_1 and could the formula be expressed as $\phi_1 = \cos^{-1}\left(\frac{C}{C+\Delta C}\cos\phi\right)$, which was just a simplification of the formula?

Dealing with Fig. 23, which showed the pitting on a second-reduction gear, he asked whether the sub-title of that figure should not have been prefixed by some such word as "expected", since he gathered that this had not yet been verified.

He noted that a value of 15,000lb. per sq. in. was quoted as a permissible bending stress on all line shaft components. This to him seemed rather high, assuming that it was nominal and neglected stress raisers, and bearing in mind that one would inevitably encounter additional bending moments due to ship movements. It would seem that where such limiting bending moment values were calculated, it would be good practice actually to measure them by strain gauge technique on the trials, which was a fairly simple matter in these days.

One important item affecting internal alignment of double-reduction gears was the low speed fine tooth coupling, which the authors had not mentioned. Presumably, this was because in most United States practice fine tooth couplings were fitted to the second-reduction pinions as a matter of course and they were assumed to provide adequate axial freedom for the second-reduction pinions and also to enable them to take up their correct attitude without restraint from the quill drive. However, in this country and on the Continent there were a few firms who preferred to bolt their quills solidly to the after end of the pinions, arguing that under torque fine tooth couplings became locked solid anyway and thereby could cause danger to the primary wheel locating bearings if such bearings had too fine axial clearances. Whilst this might be so for main wheels cut in the same direction for both helices, that is to say, with their cumulative pitch errors in phase, it might possibly not be general experience, particularly where high frequency errors were involved, such as worm error undulations, which Dorey and Forsyth* had shown could lead to very high dynamic loading. He said he would greatly appreciate the authors' views and experience on this point of design.

* Dorey, S. F., and Forsyth, G. H. 1947. "Some Gear Cutting Inaccuracies and their Effect on Gear Loads and Gear Noises". Trans.N.E.C.I., Vol. 63, p. 267.

Finally, he suggested there was a lot to be said for isolating the main reduction gears from the line and thrust shafting, as was done from the turbines. There could be little doubt that this would be kinematically desirable. If this were done, he said, the authors might also get away with a somewhat smaller computer!

As examples of what might be possible in this direction, he said he would like to quote two fairly small ships, ore carriers, classed with Lloyd's Register. They were propelled by nested geared turbines installed aft and developing about 3,600 s.h.p. at 116 r.p.m. These ships had been in service for five or six years, and both were liable, he understood, to navigate waters in which a large amount of floating debris, including logs, was present. Consequently, in order to increase the resilience of the short line of shafting against shock loading, both ships were fitted with torsionally flexible spring couplings of a well known make interposed between the main gear wheel and the separate thrust. These installations had proved absolutely reliable and trouble-free in service, so far as could be judged from the survey reports. Further, there had been no reports of gearing trouble in either ship, although it was clear from the number of reports of propeller blade damage that the service conditions had been onerous in this respect. The couplings were about 7ft. 6in. in diameter and weighed six tons, of which only about two tons were carried by the after main wheel shaft bearing. The main advantage of such a coupling was that it did not transmit bending moments and also accommodated both axial, torsional and angular deflexions. Thereby it should comply ideally with the authors' specification for limiting difference of loading on the forward and after main wheel bearings, thus ensuring that the secondary pinions meshed correctly with the main wheel. He would greatly appreciate the authors' views on such a possibility for larger and more powerful ships.

Commander A. J. H. Goodwin, O.B.E., R.N.(ret.) (Member) said that the authors were to be congratulated on a very thorough exposition of the possible effects on gearing alignment caused by the shafting and turbine connexions.

He said he would confine his remarks to the question of shaft alignment, since there were others better qualified to comment on the proper alignment of turbines to gearing.

In Naval installations they had for some years been working along somewhat similar lines in determining the best methods for lining up shafting in such a way as to minimize:

a) Adverse effects from hull distortion caused by enemy action.
b) Wear on A-bracket bushes.
c) Stern gland leakage.
d) Bending stresses in the shafting.
e) Maldistribution of gear wheel bearing loads.

The general principles that had evolved as a result were the following:

1) The line of shafting should be kept as flexible as possible by the use of the fewest bearings that would prevent shaft whirling. If these could be fitted adjacent to the bulkheads, so much the better.
2) Bearings, and particularly the A-bracket bearing, should be aligned to the natural elastic slope of the shaft.
3) A non-wearing oil lubricated bearing should take the place of the conventional stern tube bush and stern gland. This could take the form of an oil lubricated stern tube bearing with radial oil seals, or a plummer block as far aft as possible with a simple gland at the ship's skin.
4) The distance of the nearest bearing from the gearing must be a compromise, taking into account gearing main wheel bearing load distribution and bending stresses in the shafting.

He and his colleagues had, in effect, started from the opposite end from the authors, i.e. at the A-bracket rather than the gearing, and this was, in fact, the way in which their shafting was physically lined up. The procedure adopted was to establish, in the drawing stage, the preferred longitudinal position of the various bearings and then to calculate the natural elastic line of the shafting with, ideally, no bending moment or shearing stress at the shaft couplings. Exceptions might occur in some cases at the final coupling between the shafting and the gearing where a bending moment was sometimes introduced to equalize the gravity loads on the main wheel bearings and also when temporary props had to be used. The distribution of bearing loads and bending moments was then examined to ascertain to what extent adjustment of any of the bearing positions was desirable on either of these accounts.

Having established a satisfactory elastic line on paper, it was necessary to position this relative to the line of sight, joining the propeller centre to a point in the gearing room which would give gear case chocks of about the required thickness. The positions in space of the supports for the tail shaft could then be determined relative to the line of sight in the ship. The A-bracket was then bored, taking account of compression in the A-bracket bush material, the slope of the tail shaft through the A-bracket and the bearing clearance. The tail shaft was next put in place, together with the propeller or an equivalent weight. If its flange coupling was not forward of the aftermost bearing ahead of the A-bracket, a temporary prop had to be used for the shaft, so that the coupling took up the calculated position relative to the line of sight. The effect of the removal of this and any other necessary temporary props had in due course to be taken account of in the calculations. The next forward length of shaft was now offered up on its bearings, the heights of which were adjusted to make its after flange concentric and parallel with the forward flange of the tail shaft. This procedure was continued until the gear wheel coupling was reached.

The alignment of the latter was carried out with the ship waterborne, and, in determining the gap required at the bottom of the flange to give equal loading on the forward and after gear wheel bearings, account had not up to the present been taken of the thermal expansion and journal position suggested by the authors. It seemed that this could usefully be included, despite the fact that the natural flexibility of Naval shafting arrangements led to a relatively small maldistribution of load. An estimated thermal expansion of 0·011in. in a current design led to a change in load of 0·15 tons on the after bearing, where the gravity load was $3\frac{1}{2}$ tons and the full torque load about 30 tons.

The higher main gear wheel bearing loadings adopted in Naval practice, namely, 350lb. per sq. in. rather than 150 helped to restrain journal movements, and he thought it was perhaps questionable whether with flexible shafting too nice an alignment need be attempted.

Mr. J. H. Milton (Member) said he would like to thank the authors for a very interesting paper.

The pitting of second-reduction gear teeth was all too common, and reading this paper made one wonder how it was possible that any installation lined-up cold coupling to coupling, as he believed most were, could possibly operate satisfactorily. It had always seemed wrong to him that present day gears, cut as they were to such fine limits of accuracy, should by virtue of their large bearing clearances be at the mercy of external malalignment creating influences, such as line shafting, especially as in tankers.

The information contained in this paper was, he felt sure, of great interest to everyone concerned with the deterioration of gearing in service. In Lloyd's Register Engineering Investigation Department they were frequently in contact with such troubles and felt that there were in addition other causes worthy of mention.

Referring again to tankers, vibration of the after end brought about by out-of-balance thrust at the propeller could result in a vertical vibration of the whole gear box at a frequency corresponding to the propeller revolutions times the number of blades. This in a recent investigation had shown the gear box to have a vertical vibration amplitude of over 1 mm. when the wear down of the tail shaft was approaching the normal re-wooding clearance, this amplitude being reduced to negligible proportions when the shaft was re-wooded. Such vibration conditions must, they felt, result in extra wear and tear of the rotating elements inside the gear case and would indicate perhaps the advisability of considering the shape of the after body of the hull, the propeller design, and the possibility of fitting an oil gland to the tail shaft at the design stage, as once the vessel was in operation this after end vibration was very difficult to eliminate.

Very little was said of bearing clearances in this paper, and when investigating the cause of end pitting of main wheel teeth, it was often very difficult to ascertain which one, or maybe two, of three pinions in mesh was causing the trouble, especially so when the bearing clearances at each end of a pinion, whose bearing loading was on the cap, did not tally and for which no original wear-down readings were available.

It would appear that if precision boring of the gear case and positive location of the pinions and wheel could be accomplished, troubles through overloading of teeth by malalignment would be minimized. Could the authors explain why, with a view to this end, more use was not made of roller bearings. Split roller bearings were used on line shafting, and they could be obtained in stainless steel.

In conclusion, he mentioned that should the lead-in of line shafting to a gear case be suspect under hot conditions, it had been found possible to verify the existing conditions by means of a simple strain gauge technique, which did not involve any dismantling.

Lieutenant Commander E. W. Ward, R.N. (Associate) said the Admiralty considered that the authors had made a notable contribution to the problems concerned with the installation of gearing in ships, and the information contained in the paper could well help to solve some of the problems encountered in British warships.

While it was noted that the information contained in the paper was mainly theoretical, it could be the explanation of pitting that had occurred haphazardly in a class of naval ships designed about ten years ago, which, although using the same design of machinery, was fitted with gearing made by several manufacturers and installed by several different shipyards.

In addition, instances had arisen where, in the same twin-screw vessel, inexplicable pitting had occurred on one set of gearing only.

The general principles now proposed by the authors had been embodied in Admiralty specifications for new ships with the exception that no allowance had been made for relative expansion between gear cases and shaft bearings and gear cases to turbines.

While this seemed to be important so far as vertical expansions were concerned, it would appear that transverse movements could be affected by distortions of the gear cases due to torque, and this might offset those due to temperature.

Values for distortion for a typical Admiralty application had been given in a paper by Waterworth at the International Gearing Conference in 1958*, the estimated effect of the distortions on the secondary pinions/main wheel tooth contact being nine-tenths and four-tenths of a thousandth of an inch misalignment over a face width length of 22in. This misalignment did not show up in practice and might well have been partially offset by transverse misalignment of the main wheel due to unequal bearing loads imposed by incorrectly aligned shafting.

In one twin-shaft ship it had been found that when turning at high speed the main wheel forward journal of the inner shaft on the turn lifted sufficiently in its clearance to blank off the oil inlet hole in the top half of the bearing. This movement of the main wheel was assumed to be due to the precessional effect but would not have occurred to such a damaging degree if the vertical loads on the gear wheel

* Waterworth, N. 1958. "Effects of Deflection of Gears and Their Supports". Proc. Int. Conf. on Gearing (I.Mech.E.), p. 43.

journals had been more evenly distributed, as recommended in this paper.

The modern naval warships were usually of light weight, all-welded hull construction and were likely to suffer far more from hull distortion than a merchant ship. Consequently it could be argued that really accurate alignment between shafting and gear box was not so important.

However, it was the Admiralty view that as every effort was made to ensure that the internal alignment of the gear elements was correct, the recommendations made in this paper, in so far as alignment of the shafting and turbines to gearing was concerned, should be taken into account.

MR. W. BLACKLOCK (Associate Member) said that the authors had stated early in the paper (on page 136) that pitting usually occurred at the ends of the pinion teeth, but, apart from malalignment, increased tooth loading at the outer ends could also occur due to differential longitudinal expansion between pinion and wheel when running with apex leading, which was described by Archer* in the TRANSACTIONS of the Institute of Marine Engineers for 1956, and it would be of interest to hear if the authors had any evidence to offer on this subject. The appropriate compensation for this temperature effect was to arrange for apex trailing, although this might be G.E.C. practice, judging by the gear wheel shown in Fig. 1. A further aggravation of this condition was that imposed by combined torsional and bending deflexions of the pinions, which tended to concentrate tooth loading at the outer ends, particularly at the torqued end, which is practically always at the after end.

In addition, the after end of the gear case, particularly at the corners, could suffer from lack of rigidity due to what had been called the "step effect" when the seating was not carried sufficiently aft over several frame spaces.

The cumulative effect of all these conditions at the after end could influence the tooth loading at this critical position, and it might well be that the remedy lay in applying generous helix correction to the teeth as in the case of hardened gears developing high torque. In the table just below Fig. 3 on page 137, it was concluded that the pressure angle stated, viz. 20 degrees, was that in the plane of rotation and not the normal pressure angle.

In the section describing turbine to pinion alignment, it was noted that all calculations were based on the full ahead condition, with which it was agreed, but since astern running had the effect of altering the bearing reaction positions, in some cases as much as the total oil bearing clearance, particularly if all the astern power was developed in the l.p. turbine, it was essential to know the amount of malalignment flexible couplings would accept without damage. Accordingly, he asked if the authors had had any experience in this respect, either by accident or experiment.

A critical relationship with regard to flexible coupling design appeared to be the teeth P.C.D. to coupling length ratio. On a series of ships with which he had recently been associated, severe pitting and wear had occurred on the secondary gearing elements, which had led to their early replacement. The gearing was of the tandem type, driven by h.p., m.p. and l.p. turbines having short flexible couplings with a P.C.D./length ratio of approximately 1 : 1 fitted between the after end of the primary wheel and forward end of the secondary pinion.

These couplings had been aligned line-in-line in the shop without regard to temperature or bearing reaction effects in accordance with the firm's usual practice, but at the time the gears were renewed, the h.p. and m.p. couplings had shown extensive wear with little or no wear on the l.p. It was significant that only on the l.p. train were the primary wheel and secondary pinion bearing reactions in the same direction in the full ahead running condition. Among other things, inadequate articulation had been held to be partly responsible for the gearing defects and quill shafts had been fitted in place of the short, stiff couplings.

Whilst the authors did not refer to alignment between primary and secondary elements, presumably both temperature and bearing reaction effects were taken into account in the same manner as that described for turbines to pinions.

In the procedure described for alignment of turbines to pinions dealt with towards the end of the paper, no doubt some provision was made to locate the turbine rotor axially whilst sweeping the pinion flange for face-to-face dial gauge readings; in spite of this he would still prefer to check these readings with a stick micrometer.

Table XVII on page 161 was very useful for determining the temperature effects on gear cases, but its value would be enhanced if the authors were able to tell them if the theoretical thermal expansion were corroborated by actual measurement. Further, the assembly room temperature was given as 80 deg. F., and it would be of interest to learn whether hobbing, post-hobbing treatment and gear case boring were all carried out at the same temperature, since only one gear manufacturing firm, to his knowledge, carried out this practice, together with assembly, all in one temperature controlled shop.

One point which he felt bound to make as a classification society surveyor was that when firms deliberately malaligned shafting in the cold condition, instructions should definitely be placed on board, preferably stamped on a plate permanently fitted to the turbine or gear case and not merely in the chief engineer's data book, which sometimes went astray after a series of changes in personnel. These instructions should also include draughts forward and aft, boilers full or empty, on or off, and the capacities of the tanks adjacent to the machinery.

Finally, he felt that this paper provided them with a much clearer picture of the actual alignment under working conditions. He suggested that perhaps they were now reaching the stage where alignment problems were fully understood by the marine engineer and any further development on the subject should be left to the naval architect to give them guidance regarding structural flexibility.

MR. J. H. GOOCH, M.A., said he would like to join with the other speakers in expressing his thanks to the authors for an interesting and stimulating paper. It was particularly gratifying to learn more about this subject from their friends from the other side of the Atlantic.

He agreed entirely with the authors that bull gears did slew in their bearings. His firm provided bearing clearances of 0·015in. to 0·020in., which allowed plenty of room for this slewing to take place. He agreed, too, that the line shaft applied a moment to the bull gear and this moment would certainly tend to make the bull gear slew in its bearings. He said he would like to mention in passing that primary wheels in double-reduction gear boxes were also liable to slew in their bearings. This was due to the fact that it was common practice in Britain to bolt the quill shaft solidly to the primary wheels, thus causing an overhung weight on each primary wheel, which would result in the wheels tending to slew in their bearings.

He was sorry to say that he had some doubt as to whether this slewing of wheels could be avoided or even mitigated by prior calculation or careful design. The extent of the slewing depended upon the magnitude and direction of the couple on the main wheel applied by the line shaft. This was in turn dependent upon the relationship between the line shaft bearings and the gear wheel bearings. This relationship was affected by the deflexion of the ship's hull and also the deflexion of the gear box. The deflexion of the ship's hull could be appreciable, and Mr. Bunyan had already mentioned a figure of 1in. per 100-ft. run. In a cargo ship, deflexion of the hull changed appreciably according to the loading of the ship, whether it was unloaded, partly loaded, or fully loaded. If it were only partly loaded, then the distribution of the cargo in the holds also had an effect on the deflexion of the hull, and therefore upon the line shaft bearings' relationship

* Archer, S. 1956. "Some Teething Troubles in Post-war Reduction Gears". Trans.I.Mar.E., Vol. 58, p. 309.

to the bull gear bearings. Changing temperature in the gear box, and consequent expansion or contraction of the gear box, would also alter the bull gear bearings' relationship to the line shaft bearings. Finally, there was the question of the movement of the ship in rough water, which caused continual changes in the relationship between line shaft bearings and the gear box.

For these reasons, his firm had had their doubts for a great many years as to whether the moment applied by the line shaft on the gear wheel could really be calculated for all the many conditions to which the ship was subjected.

He said he would like to make at this point a short digression into history. It was now exactly fifty years since Sir Charles Parsons put to sea the first merchant geared turbine ship, which was the *Vespasian,* in 1909. As soon as Parsons decided to have a gear box between the turbines and the propeller shaft, he was faced with the problems that might arise from this slewing of the bull gear in its bearings. At this time, a solution to this imagined problem was proposed by two engineers, named McAlpine and Melville, who designed in 1909 a floating frame to support the pinions in the gear box. Although Parsons decided to try out the floating frame, his first gear box without the floating frame worked very satisfactorily and so he decided to continue without using floating frames. A few years later, in 1916, an engineer named Alquist advocated to Parsons and others the use of gear wheels of unusual design. The wheels were built up of a large number of discs. The idea behind this design was to enable an even distribution of load to be obtained over the face width despite the slewing of the gear wheel in its bearings and despite any pinion deflexion. Parsons considered this, but decided not to use this multiple disc wheel and stuck to his solid wheels. These multiple disc wheels were subsequently adopted in America, where they were manufactured for a number of years in the 1920's.

Towards the end of the first world war, there was a final attempt made to counteract the effects of bull gear slewing in their bearings by supporting the pinions in bearings, which were in turn supported by springs, thus allowing the pinion to swing itself to counteract any alteration in the alignment of the bull gear. Parsons tried this idea and manufactured gears in which the pinions were so supported, but discovered that the gears performed no better than without this elaborate addition and he dropped it in subsequent designs.

By that time it was about 1919 and nothing further had happened in this particular subject until their study of it today.

He said that his firm had the experience of putting a great many gears to sea and undoubtedly many of them operated with their gear wheels slewed appreciably in their bearings, and yet they still found that a good proportion of them had good contact markings across the entire face width of the gears, despite the slewing of the wheels.

He felt that an explanation was now required as to what was the attitude towards this very important subject. They were inclined to believe that the deflexion of the gear box itself had a substantial effect and that they were very conscious of the fact that the gear box was an elastic body. If there arose, due to the bull gear slewing in its bearings, a concentration of load at a particular end of the face width, say, for example, at the forward end, then the forward pinion bearing would have a larger bearing reaction than the aft pinion bearing. This would naturally deflect the forward bearing more and thus tend to restore to a certain extent the loss of alignment which was caused by the gear wheel slewing. This could only be considered as likely to happen if it could be established that the deflexion of the pinion bearings was of the same order as the amount of slewing of the bull gear in its bearings.

There had been very little work done on this most important subject until recently, but Mr. Waterworth* presented a very interesting paper on this subject at the International Conference on Gearing held under the auspices of the Institution of Mechanical Engineers in September 1958. He had the author's permission to reproduce an illustration from his paper (Fig. 39). The figure showed that the deflexions of the pinion bearings in the gear box examined were appreciable and of the order of 0·010in. to 0·012in.

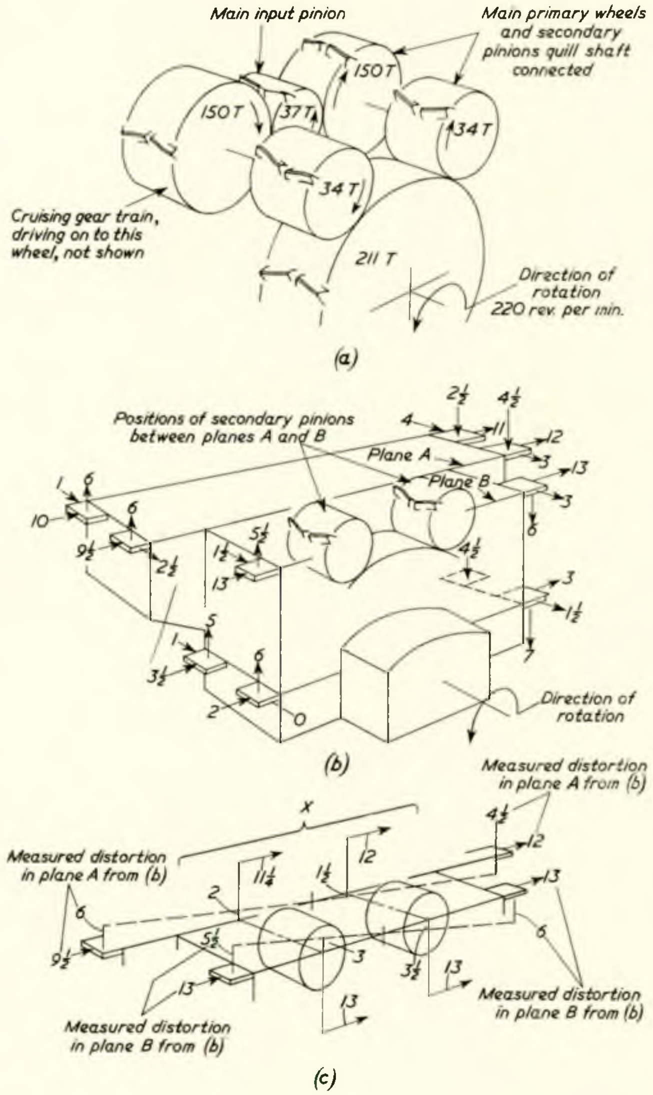


Fig. 39

Experience had taught them that it was unlikely that one could actually prevent bull gears slewing in their bearings, and they were inclined to believe that, even if one could, it might not result in a substantially improved gearing performance. This, however, did not detract from his admiration of this paper and of the authors' work, which he found most interesting. He hoped that it would stimulate further interest in the subject which he felt had been rather neglected during the last forty years.

Mr. R. E. Salthouse, speaking also on behalf of Dr. J. F. Shannon, said the authors had done a great service in emphasizing the importance of alignment of line shafting and the main wheel.

The problem had been greatly exaggerated by the use of too many bearings in the line shafting in many ships, and this was confirmed by the authors' examples. The tanker quoted had an L/D ratio (L being the length between the bearings and D the diameter of the shaft) for the span

* Waterworth, N. 1958. "Effects of Deflection of Gears and Their Supports". Proc. Int. Conf. on Gearing (I.Mech.E.), p. 43.

between the main wheel and the first line shaft bearing of 7·2 : 1. On each intermediate shaft the ratio was 7 : 1. For the bulk cargo ship the ratio for the first span was only 4 : 1. The authors showed that alignment conditions were greatly improved by removing the first line shaft bearing when the L/D ratio became 12·5 : 1.

Very much greater spans were used in some naval vessels, where ratios of 30 : 1 were used between bearings and the ratio of 20 : 1 might apply between the main wheel and the first bearing. This relieved the problem immensely.

With these long spans, there was still a safe margin against whirling speeds and bending stresses. The whirling speed, neglecting axial thrust, was $4 \cdot 24 \times 10^6$ (L/D^2) r.p.m., the corresponding bending stress due to sag being $\pm 0 \cdot 283$ (L/D^2) lb. per sq. in. Thus the criterion to be considered was the (L/D^2) factor. If an allowance was made for fatigue, a maximum permissible bending stress as low as ± 1 ton per sq. in. was permitted. Even with this limitation, the spans for a 10-in. diameter shaft would be 23·5ft. long and for a 20-ft. shaft it would be 33ft. long, the L/D ratios being respectively 28·2 and 19·8 and each gave a whirling speed of about 535 r.p.m. Why, in these circumstances, did they persist in using short spans?

The use of longer spans relieved the problem on the gear, but it presented a minor difficulty in aligning what was more or less a flexible shaft. As shown by the authors, this difficulty could easily be overcome by raising the line shaft bearing nearest to the gearing by the amount required to carry the weight of the overhanging portion. It was not suggested, however, that the spans near to the propeller shaft should be extended, because there very different conditions applied.

The use of the digital computer was now universal, but the engineer normally checked a result by as close an approximation as possible. In the present example, this could be done fairly accurately by considering the line shaft as a propped cantilever, "built in" at the second bearing aft of the main wheel and free at the main wheel coupling. This checked fairly well with the full calculation and was a valid approximation, because, as shown by the authors, the after bearings had very little effect upon the results.

It was noted that the authors took the position of the journals in the bearings without any reference to the effect of oil films on journal attitudes. In a double-reduction, single-helical gear unit, the effect of the attitude and eccentricity of the journals in their bearings was investigated by the contributors. Using experimental data from bearing tests, it was found that the openings of the teeth were greater than those due to bending and twisting. A review of the situation, taking secondary effects into account, led to some doubt as to the reliability of the analysis. As the gear was to be thoroughly investigated on the test bed, it was decided not to apply the predicted helix modification but to await the results of the trials. So far, the evidence of the trials was that correction for attitude was not necessary. He wondered whether the authors had made similar investigations which had led them to ignore the effects of oil films and bearing attitudes.

Correspondence

Mr. A. Cogman wrote that for some years past it had been the practice in the Engineering Investigation Department of Lloyd's Register of Shipping to check shafting alignment by a method which enabled the state of alignment to be determined without breaking couplings. This technique had not been developed to supersede existing methods when aligning new shafting; but rather to enable a rapid check to be made when shaft alignment in an existing vessel became suspect.

The method basically involved measuring the bending strain in the line shafting at intervals along its length. This was done by fitting a strain gauge at the desired position and rotating the shaft by means of the turning gear. In this way the bending moment existing at the point of attachment of the gauge could be evaluated. By repeating the measurement at suitable points, a bending moment diagram might be constructed, and by choosing measuring points at which the stress due to the shaft weight was zero, such a B.M. diagram would represent the degree of misalignment or otherwise of the shafting.

The above description was necessarily brief and in practice special consideration must be given to the end conditions at the main gear and propeller shaft; additional measuring points might be called for here.

Interpretation of the B.M. diagram was most easily accomplished by means of a double graphical integration which then provided a deflexion diagram.

Instrumentation for the method was originally by means of an electro-mechanical strain gauge which could be attached quickly to any point on the shaft; electric resistance strain gauges were considered but at the time (1950) cements in use were slow drying and this would have unduly protracted the time required.

More recently, very rapid setting cements had been developed, and present practice was to use electric resistance strain gauges and such a rapid setting cement as, for example, "Eastman 910". An installation at as many as 12 points on a shaft could be attached, wired and ready for use in about two hours.

As stated at the introduction, this method of assessing shaft alignment did not in any way replace existing methods of checking when installing shafting; it provided, however, a valuable means of checking the alignment of suspect shafting in a vessel in service, with a considerable saving in time over other methods.

One point that might be worthy of consideration was the use of the strain gauge method in assessing the difference between hot and cold conditions. Here the time saving features of the technique would enable readings to be taken immediately after shut-down and such readings could then be compared with those taken from the cold installation as originally set up.

Dr. A. W. Davis (Member) was aware of the reputation which the authors enjoyed in the United States and of the renown in which their company was held for the quality of the marine reduction gears they produced. The elegance with which the paper under discussion had been produced testified to both these facts and in these circumstances the authors' recommendations could neither be condemned so heartily nor discarded so lightly as might be prompted by experience in this country that was contrary to the experience described.

At a verbal discussion which followed the presentation of the paper in Scotland, he had the impression that he had quite failed to convince the authors that it was his experience that full length contact on the secondary (as well as the primary) gears was maintained in service with line shafting that had been set up line-in-line with the thrust shaft and gear wheel shaft in the cold condition. It also appeared to surprise the authors to learn that it was the writer's experience that the complete absence of wear on the teeth after many years of service meant that the truth of running alignment could only be checked by very close examination indeed of the tooth surfaces. In the light of the authors' remarks, this condition had just been examined in a vessel 12 years' old and the writer's experience was firmly substantiated in that there was still evidence of light shaving marks along the whole length of the teeth. The authors did not in fact describe how they assessed the condition that only the forward two-thirds of the secondary gear helices bore the load when set up line-in-line with the shafting when cold. Was the effect witnessed by distress of the tooth surfaces, or were the gears cleaned and marked with lacquer when in the hot condition and again run in that state without being allowed to cool down? This would seem to be a difficult series of actions to carry through in view of the need for having the tooth surfaces dry and clean while the lacquer was applied and allowed to harden.

As a substantial part of the authors' case depended on merchant gears, and having regard to the similarity of Classification Society allowable loadings, it seemed improbable that the difference between the authors' and the writer's experience should lie in gear loading *per se*. One searched for some essential mechanical differences in practice, but, apart from the alignment itself, the only difference that the writer had observed, appeared, firstly, to lie in the oil drainage arrangements in the gear case, it being usual practice in this country to fit a sump that had the incidental effect of preventing hot oil from heating the gear case seatings. A rapid check aboard many a British built ship steaming at full power would show that vertical expansions producing main wheel bearing displacements of the order referred to by the authors did not exist.

A second difference was the American practice of adopting welded rim construction, giving lighter main wheels than either bolted, or more particularly shrunk, construction, with cast iron centres and hence wheels that are more likely to be influenced in their rotational position by forces arising from irregularities of alignment.

On the theoretical side of their examination the authors had disregarded the rotational effect of the main gear wheel in its effect on the attitude of the shaft in the bearings. From the results of bearing research carried out at Pametrada it was the experience of the writer's firm that in the type of gearing under discussion there was virtually no lateral displacement of the gear wheel shaft under running conditions. In British practice the amount of vertical rise on the oil film was perhaps minimized by the provision of a finer oil clearance than was referred to by the authors.

Some of the authors' examples referred to very large gears and this was a fact that should perhaps be borne in mind by those who in the future would always be looking for the first slight evidence in support of the authors' remarkable contentions.

MR. J. HICKS would be interested to know if the authors had any records of the bending moments in the intermediate shaft or thrust shaft in the hot operating condition on installations, in which their method of alignment had been used.

Whilst every possible accuracy had been attained by the authors in calculating the expansion of the gear box both horizontally and vertically, no reference had been made to the distortion of the gear box due to torque. The writer would be interested to know the reasons which led the authors to disregard this fact.

In view of the authors' remarks regarding the large permissible wear-down of stern tubes and the influence of this wear-down on the intermediate and bull gear bearing reactions, the writer was of the opinion that a review of the conditions imposed on tail shaft bearings was now due, with a view to the use of lubricated white metal bearings with characteristics similar to those of the intermediate shaft bearings.

MR. H. NILSSON considered that this useful paper proved the fact that it was of little use to increase efforts to improve the design and fabrication technique of propulsion gears if equal attention were not paid to the installation of the gears and the special conditions under which they had to operate.

He had the following comments to make regarding the effect on the bull gear by the line shafting:

1) The authors had shown that an external bending moment in the shafting would produce a misalignment between the bull gear and its pinions. They had pointed out that this was due to unequal bearing reactions on the bull gear journals. These reactions would displace the journals unequally and force the bull gear into a skewed position.

   However, the non-symmetrical deflexion of the bull gear shaft, which resulted from the unequal reactions, should also be considered. It could be shown that this effect would tend to incline the bull gear and decrease the misalignment to a certain extent.

2) The authors had assumed that the thrust bearings did not absorb or introduce any bending moment. In other words, the bearings were of the self-aligning type. If the non-self-aligning type were used, an additional bending moment would be imposed on the shaft and the misalignment of the bull gear would be increased. The moment was produced when the thrust bearing foundation was deflected by the thrust. The thrust pads would then give an unequal load distribution on the thrust collar. The bending moment which was introduced in this way could be several times as large as the other moments in the shaft even if the thrust bearing were rigidly connected to the ship structure. Thrust bearings of this type could consequently have an adverse effect on the alignment of the bull gear.

Authors' Reply

The authors sincerely thanked each of the contributors to the discussions, and the companies they represented, for having so greatly enhanced the value of the paper by their well-considered comments.

Both Mr. Peach and Mr. Engvall mentioned the possibility of the pinions shifting in their journals to accommodate the skewed bull gear. From calculations on a few gear units, they found that the original calculated mismatch that was based on uniform tooth loading was reduced approximately 10 per cent due to the compensating or restoring forces on a mating pinion and bull gear, which acted when parts were in a misaligned condition. For example, if on a given gear unit a 20,000-lb. unequal load on bull gear bearings produced a calculated mismatch of 0·0005in., disregarding any compensating effects, then the final result when the pinion and gear shifted slightly in their clearances, due to the non-uniform load applied to them, would be that the 0·0005-in. mismatch was reduced to approximately 0·0004 to 0·00045in.

Since the 0·0002-in. mismatch allowance was in a sense an arbitrary guiding limit, at this time their practice was to neglect the compensating effect when calculating the allowable unequal load on forward and aft gear bearings. They had not included the flexibility of the teeth as cantilevers in the calculations, which, if considered, would also tend to increase the calculated allowable mismatch limit. More study and experience, of course, might change this practice.

The 0·0002-in./ft. limit, however, was established by observations of tooth contact distribution under controlled conditions on many ship propulsion gear units in the factory under partial and full load tests. Actually, therefore, the compensating effects mentioned above, and others, if they existed, had been taken into consideration. They knew that if this value were exceeded to any large degree on past and present designs of ship propulsion gears, uniform tooth contact in the face width would not be obtained, and the teeth would be more heavily loaded at one end of the helices, to such an extent that surface failure would result eventually. A detailed theoretical analysis, of course, should be made more firmly to establish and substantiate the mismatch limit.

They must point out that shortly after they began their concerted study of tooth misalignment about two years ago, the most important event that happened and encouraged them to continue the study was when they witnessed the actual tooth contact distribution on a gear after sea trial, and it agreed with predictions based on calculations.

This particular gear was carefully installed, and accurate checks were made during assembly in the factory and again in the ship, to establish and show that uniform tooth contact distribution existed. After dock and sea trials, a non-uniform tooth contact condition showed up in the second-reduction mesh—only partially across, and heavy toward the aft end on one of the second-reduction pinions, but practically all across and slightly heavier toward the forward end of the other second-reduction pinion.

From previous experience, the observation of these contact distributions indicated that the worst affected pinion was off alignment in its tooth mesh approximately 0·0006in./ft. of face width, and would require approximately 0·004-in. removal of babbit from the bearing reaction area in order to correct it. The contact of the other pinion agreed with a mismatch of approximately 0·0002in./ft. or less.

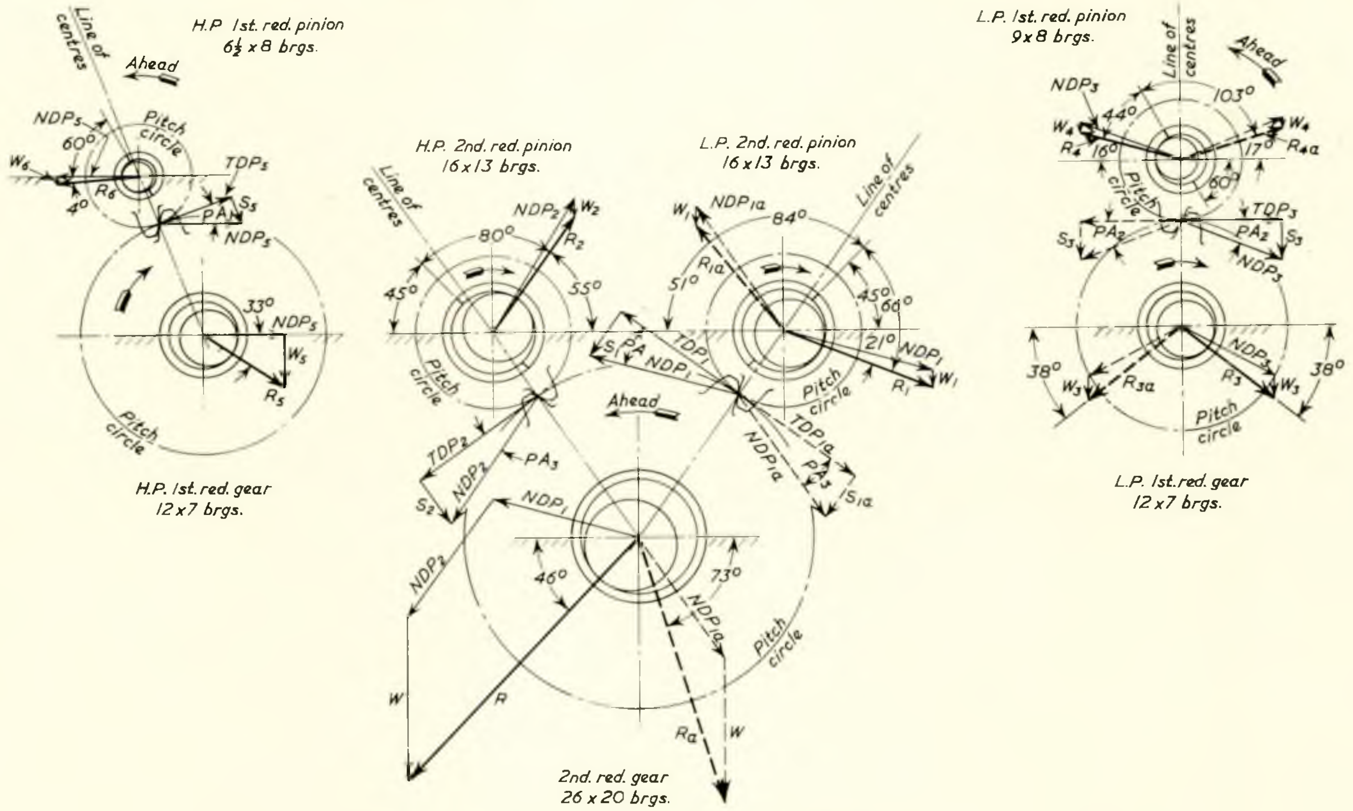
The observed tooth contact distributions on respective pinions agreed with the direction of misalignment calculated, if the heavier load were on the aft second-reduction gear bearing. Subsequently, the bearing reactions on this gear unit were measured using calibrated jacks, and the loading was actually found to be heavier on the aft bearing. Also, the difference in loading on gear bearings was found to be of a value such that the calculated mismatch agreed approximately with the amount of non-uniform tooth contact distribution actually observed. In this particular gear unit it was decided to correct the misalignment by scraping bearings and re-chocking the corner of the gear casing; and here again, the corrections that were necessary and were finally made agreed with the calculated amount of bearing scraping required to obtain uniform tooth loading.

They had since witnessed many examples of non-uniform tooth contact on other ships, which substantiated that qualitatively, and to some extent quantitatively; they could calculate how gear to line shaft alignment affected the internal tooth alignment. It was going to take more time, study, and experience to establish definite proven limits of mismatch for all types of propulsion gearing, that everyone would believe and accept without question. In the meantime, they felt that the method of calculating outlined, and the 0·0002-in. limit established, gave unequal bearing load limits that were reasonable and should easily be met by proper line shaft design and installation.

In answer to Mr. Peach, the anticipated effect illustrated in Fig. 23 had in fact now been demonstrated, and was obtained by realigning the line shaft to obtain more equal static gear bearing loadings. On a sister ship, which also had a tooth contact distribution as in Fig. 22, but where realignment of the line shaft was not made, it was necessary to reduce the wall thickness of the aft l.p. second-reduction pinion bearing in the ahead reaction by machining 0·006in. of babbit from the pinion bore in a radial direction. This method also produced uniform tooth contact distribution, and from experience would give satisfactory operation, even though the mating pinion and gear were still operating in a crossed-axes condition. Although scraping the bearings had been, and could be, used to correct misalignment, it was felt that in the future this should not be done, when it could be shown that the misalignment was due to an improperly aligned line shaft.

Mr. Engvall mentioned that a more serious reason for misalignment in second-reduction mesh could be the internal misalignment caused by insufficient flexibility between first-reduction gears and second-reduction pinions; and stated that the direction of movement of the first-reduction gear and second-reduction pinion, due to load, was not in the same direction. They were not entirely in agreement with this statement, and would explain briefly their viewpoint.

Fig. 40 was a bearing reaction diagram of a typical gear for a tanker or cargo ship, when operating at full power ahead and looking in the aft direction. It showed where the journals "rode" in their respective bearing bores by means of the total resultant force vectors, *R*. They would note that the high



Solid lines: forces with ahead rotation
Dotted lines: forces with astern rotation

FIG. 40—*Bearing reaction diagram*

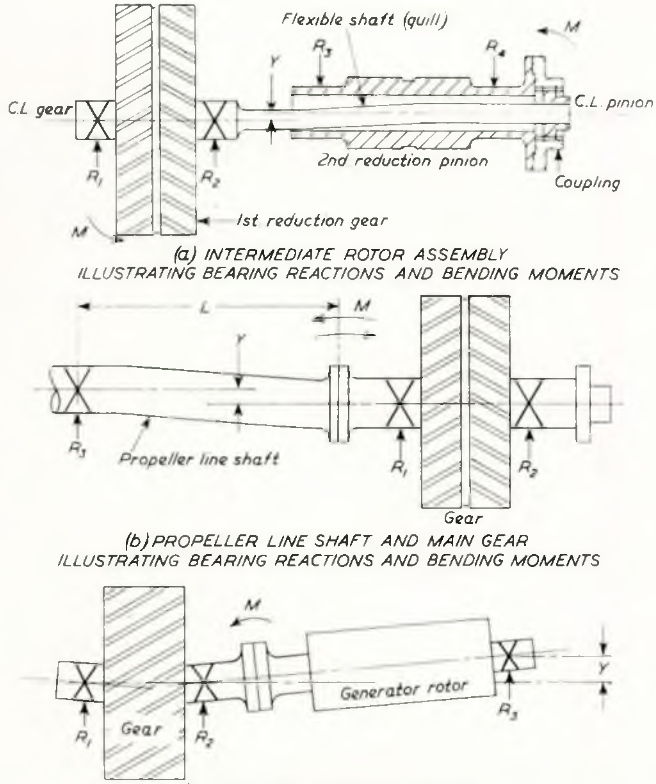
18,750 h.p. ahead power (9,375 h.p., 9,375 l.p.); 6514/3400/772/106 r.p.m.
TDP = tangential driving pressure; *NDP* = normal driving pressure (plane of rotation); *PA* = pressure angle (plane of rotation); *W* = weight (due to gravity); *R* = resultant force of journal on bearing (bearing reaction is equal and opposite to *R*); *S* = separating force.

		Per bearing Lb.		Lb.	Bearing pressure, lb./sq. in.
(*TDP*)	1 =	33,767	R =	80,625	155
(*TDP*)	2 =	33,767	R_a =	70,000	135
(*TDP*)	3 =	11,560	R_1 =	37,000	178
(*TDP*)	5 =	8,520	R_{1a} =	32,000	154
	W =	38,400	R_2 =	32,800	158
	W_1 =	3,700	R_3 =	14,700	175
	W_2 =	3,700	R_{3a} =	14,000	167
	W_3 =	4,752	R_4 =	12,100	168
	W_4 =	702	R_{4a} =	11,600	161
	W_5 =	6,150	R_5 =	11,050	132
	W_6 =	475	R_6 =	9,150	176

pressure first-reduction gear journals rode at 33 degrees below the horizontal joint and inboard, as shown by R_5, whereas the high pressure second-reduction pinion, which was driven by the gear, rode 55 degrees above the same horizontal joint and inboard, as shown by R_2. Since the four bearing bores (two for the gear and two for the pinion) were all bored on the same axial centre line, and these bearings could have in the order of 0·018 to 0·024-in. diametral oil clearance, the axial centre line of the pinion journals could be offset approximately 0·020in. with respect to the axial centre line of the gear journals.

This 0·020-in. offset, Fig. 41, was represented by dimension *Y* in sketch (*a*). It could be noted that if the single engagement coupling at the end of the pinion were solid to the shaft (not flexible as shown), then the shaft would have to bend as shown for tooth alignments to be maintained at respective first-reduction and second-reduction tooth meshes. Of course, bending moments *M* would be created that would affect the bearing reactions R_1, R_2, R_3, and R_4 to some extent, so that both the second-reduction pinion and first-reduction gear might be operating at crossed axes with their respective mating parts. The bending moments would be reduced significantly by the use of the single-engagement coupling, and further reduced by the use of a double-engagement coupling, as illustrated by Mr. Engvall. However, calculations and operating experience showed that the bending moments which resulted with the use of a flexible shaft and a single-engagement coupling as used in the C-2 and C-3 designs were not significant, and the possibility of misalignment in first or second-reduction meshes from this cause was not serious.

Referring again to the bearing reaction diagram, Fig. 40, it should be noted that the low pressure first-reduction gear



(a) INTERMEDIATE ROTOR ASSEMBLY ILLUSTRATING BEARING REACTIONS AND BENDING MOMENTS

(b) PROPELLER LINE SHAFT AND MAIN GEAR ILLUSTRATING BEARING REACTIONS AND BENDING MOMENTS

(c) GEAR AND GENERATOR ROTOR ILLUSTRATING BEARING REACTIONS AND BENDING MOMENT

FIG. 41

journals rode at 38 degrees below the horizontal joint and outboard, as shown by R_3; and the low pressure second-reduction pinion rode 21 degrees below the same horizontal joint and also outboard, as shown by R_1. Since these resultant forces were very nearly in the same direction, there would be practically no bending moment present in a shaft connecting the two parts, and hence no misalignment from this cause.

It should be noted that in the typical gear as shown, it was the low pressure second-reduction pinion that was worst affected by a given amount of unequal gear bearing loading; whereas it was the high pressure second-reduction pinion that was worst affected, due to its connexion with its respective first-reduction gear.

Mr. Peach and other contributors suggested that they should establish the allowable difference in vertical bearing reactions as a percentage of the total gear weight. Since the writing of this paper, they had calculated many more gear units, with second-reduction gear weights ranging from 15,000 to 95,000lb., and found that on an allowable misalignment basis of 0·0002in./ft. of helix, the allowable unequal load was 13 to 22 per cent of the total gear weight on the gear units investigated.

Regarding the allowable forces in the horizontal direction, values in the order of 1,000 to 5,000lb. were specified as a limit, based on the study of only a few gear units. This was a difficult specification to stipulate, because if the horizontal misalignment was in such a direction that it compensated for a given unequal load condition, even larger horizontal forces could be tolerated. If, however, the horizontal misalignment was in such a direction that the forces produced were additive to the unequal loading in introducing misalignment to the gear, then the horizontal forces should be limited to small values.

In general, the flexibility of the line shaft pedestals and supports was relatively large in the athwartship direction. Therefore the tolerances on forces in a horizontal direction could most likely be greater than given in the foregoing, if the spring constants of the bearing supports were considered. On the other hand, the problems associated with obtaining a given alignment in the horizontal direction were usually not as great as those in a vertical direction.

Regarding the minimum bearing reaction of 1,000 to 2,000lb., the intention was that, regardless of the size of the bearing or its maximum load capacity, it should have some minimum specified amount of loading, so it was known that it acted as a support and provided stability to the system as intended. Perhaps this minimum value should be greater.

It was true that no matter where the first line shaft bearing was placed, it would affect the loading on the gear bearings to some extent and thus the alignment of the gear; however, this effect could be minimized by selecting a greater span, as illustrated by removing the number 3 bearing in Fig. 21. Of course, the longer span would increase the load on the after gear bearing, if all bearings were allowed to remain on a straight line; but by lowering the gear bearings or raising the line shaft bearings, the forward and aft gear bearing loading could be equalized without any significant increase in gear bearing loadings at full power.

Allowable bearing wear certainly could have an effect on shafting alignment. It appeared to the authors that the allowable bearing wear limits mentioned for gear and line shaft bearings and stern tubes must be reduced in certain cases in order to accomplish a satisfactorily operating gear and line shaft. In this respect, it should be noted that the Buships Manual stated that, "clearances and weardowns specified are to be used only in the absence of other instructions".

They knew from experience that gear bearings did not and must not normally wear down anywhere near the amounts shown in Table XX, or problems more serious than just a difference in gear bearing reactions would be experienced. Once a bearing started to "wipe" or wear down, say in the order of 0·005in., its oil clearance was destroyed, it would have an excessive temperature rise, and it would usually continue to wipe and wear down at a fairly rapid rate. Also, in most cases, the two bearings supporting a gear would not wipe or wear at the same rate; and if one exceeded the other by more than 0·003in., a serious internal misalignment was directly introduced.

They were not familiar with the rate of wear and total amounts of wear experienced on line shaft or stern tube bearings; however, it was quite obvious that if they were allowed to wear to the limits shown in Table XX, very significant changes could occur in bearing reactions.

On ships with relatively short line shafts, where the first line shaft bearing was a forward stern tube bearing that was allowed to wear down a relatively large amount, it certainly became an unrealizable goal to maintain the 0·0002in./ft. mismatch limit. As much as 0·0004 or 0·0006in./ft. of face width might be experienced that could cause tooth surface failure; and when the mismatch exceeded 0·0010in./ft., a serious tooth breakage problem could result. A ship of this type had no means of readjusting bearing positions except by reboring stern tubes or rechocking the gear unit, which of course was a very costly and lengthy operation. In some cases, a babbited spring bearing had been installed in place of the forward stern tube. The wear-down permitted for a spring bearing approximated more nearly to the magnitude of wear-down that the shaft system could tolerate, and it also provided a means of adjustment of bearing position to compensate for bearing wear-down.

On the other hand, ships with relatively long line shafts, having more than one babbited line shaft bearing aft of the gear unit could tolerate a greater amount of bearing wear-down if the line shaft bearing positions were adjusted periodically to keep bearing reactions within previously established safe minimum and maximum values.

Mr. Francis recommended that minimum and maximum bearing loadings be established, and line shaft bearing pedestals designed to allow ship's force personnel and/or repair yard personnel to adjust readily the bearing loadings. They felt this was a good, practical answer to many line shaft problems.

Mr. Atkinson, of Buships, suggested the use of strain gauges to overcome possible excessive misalignments. At this time, they felt this method was more for laboratory use, and although it might serve a useful purpose during the original sea trials, the fact still remained that a relatively simple, accurate means should be available actually to reposition the bearing pedestals as required, so that acceptable bearing loadings prevailed.

Mr. Engvall questioned whether or not the assumption that all line shaft journal centres were in a line and the centre of the free-hanging flange was in line with the journals and square to the line, could be achieved and maintained. He mentioned that, because of various conditions that could cause relative movements, the tolerances which should accompany gap and offset settings could represent a good percentage of specified values. This could be true on line shaft arrangements wherein the specified gaps and offsets were of relatively small amounts, but not necessarily true on other line shaft arrangements requiring relatively large gap and offset settings.

The authors could not agree completely with Mr. Atkinson of Sun Shipbuilding that "data under operating conditions are required on the actual position assumed by the bull gear shaft in its bearings before the indicated calculations can be accepted as reliable".

They felt that the major amount of time, effort, and money should be spent on calculations and measuring techniques that would accurately and quickly establish optimum bearing positions and loadings. Determining the position of journals in their bearings while under operating conditions required rather elaborate laboratory techniques and instruments, and if the results showed that the bull gear was skewed in its bearing clearances, then there were questions that still remained, such as: What forces caused the skewness? What were the static bearing loadings at final alignment with a ship waterborne? and were the bearing loadings measured accurately?

It had been mentioned that several factors pertaining to the deflexion of a ship's structure might have an important influence on alignment, such as the relatively large movements experienced when comparing checks on shaft alignment taken on a cool morning with those obtained in the hot afternoon sun; and also those due to varying ballast conditions encountered in everyday operation.

Because they had experienced rather large deflexions, many shipowners and shipbuilders naturally questioned any recommended shaft alignment changes which involved only a few mils.

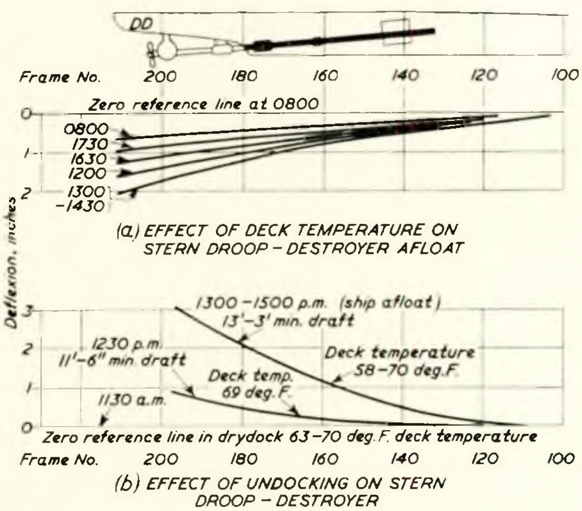


FIG. 42—*Ship deflexion due to heating and undocking*

They believed the answer to this question was found in the studies made by the Boston Naval Shipyard, wherein they measured how a ship's hull, in way of shafting, responded to influences such as temperature, load, and speed changes. In the paper to which they had referred in their Acknowledgements, "The Alignment of Main Propulsion Shaft Bearings in Ships", by Kosiba, Francis, and Wollacott, it was illustrated (by Fig. 42) how temperature variations caused the hull girder of a destroyer-type ship afloat to deflect 1½in. from midship to main strut and undocking caused a deflexion of 3in. in the same distance. Fig. 43 illustrated how speed changes at

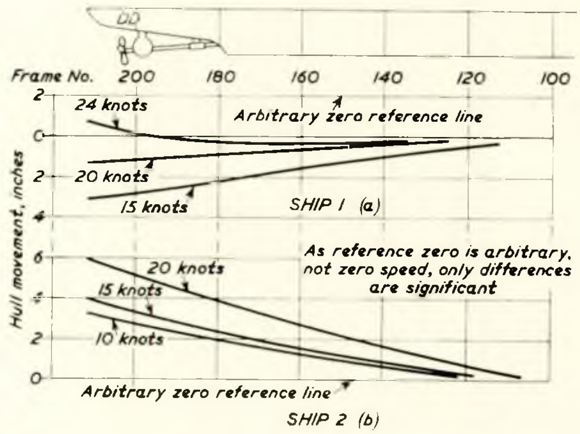


FIG. 43—*Movement of stern with changes of speed*

sea caused a deflexion of approximately 3in. It was concluded by them that the hull deflected as a fair curve, and that a straight line of bearings on a flexing hull girder was in reality a family of faired curves at different instants of time. Therefore, bearing movements relative to the hull, caused by temperature variation, changes in hull loading, or speed of ship, were not erratic, nor characterized by a randomness such as those produced by inaccuracies in installing, bearing wear, and so on. This must be true, or line shafts would not operate nearly as well as they did.

To illustrate in another manner, let them start with the condition that the sections of line shaft were disconnected, as they were when an initial alignment was being made. Now, if the bearing pedestal positions had been established in some manner, and measurements of gap and offset taken at one instant of time, say in the middle of the night, then when the readings were repeated at some other time, for example after a temperature or ballast change, the gaps and offsets between the various shaft sections would have changed considerably. This, of course, would lead one to believe that the alignment of the shafting had been seriously affected. However, it was not as serious as it might seem when it was understood that if the shaft sections were all connected together, and the ship went through the same temperature and/or ballast changes, the *bearing reactions* would *not* change significantly. This was due to the fact that the bearing positions were not changed erratically, but lay on a fair curve, conforming to the deflexion of the hull.

Because large changes in deflexion of a ship's hull could occur while taking boresight or gap and sag measurements, it was obvious that the relative movements and tolerances could be minimized by establishing procedures of alignment such that measurements and settings of gap and offset between relative shafts were made quickly, with all conditions, such as ballast, deck temperature, and the like as nearly constant as possible. After gaps and offsets had been established as accurately as possible, to calculated values and tolerances, then the alignment could be checked by direct measurement of bearing loads, using calibrated jacks where shafting was accessible for jacking.

In conclusion, and quoting directly from the aforementioned paper: "It should be appreciated that each shaft arrangement imposes its own requirements for accuracy in the positioning of bearings. It is well to restate that the bearings, when positioned on a faired curve, will normally accommodate considerable movements of bearings (without significantly changing bearing reactions), as long as they result in another faired curve. However, random departures of single bearings, resulting in a significant departure from the faired curve, can result in unloading one bearing and overloading adjacent bearings".

The authors would have liked nothing better than to have answered every question raised in both the New York and London discussions, but pressure of work had made this impossible. They hoped therefore that contributors to the London discussion would find the answers to some of their questions in the published reply, which was intended to cover both discussions.

Annual Dinner

The Fifty-seventh Annual Dinner of the Institute was held at Grosvenor House, Park Lane, London, W.1, on Friday, 11th March 1960 and was attended by 1,194 members and guests.

The President, Sir William Wallace, C.B.E., LL.D., was in the Chair.

The official guests included: His Excellency Baron Adolph Bentinck, The Netherlands Ambassador; His Excellency Senor Victor Santa Cruz, The Chilean Ambassador; The Right Honourable Ernest Marples, M.P., Minister of Transport; The Right Honourable The Lord Carrington, K.C.M.G., M.C., The First Lord of the Admiralty; The Right Honourable The Viscount Simon, C.M.G., Chairman, The Port of London Authority, and President-elect; Sir Gilmour Jenkins, K.C.B., K.B.E., M.C., Past President; The Right Honourable The Lord Winster, P.C., K.C.M.G., President, The Merchant Navy and Air Line Officers Association; Sir Victor Shepheard, K.C.B., Director, The British Shipbuilding Research Association; Sir James A. Milne, Chairman, The British Shipbuilding Research Association; K. R. Pelly, Esq., M.C., Chairman, Lloyd's Register of Shipping; W. G. Agnew, Esq., C.V.O., Clerk of the Privy Council; Commodore J. Plunkett-Cole, R.A.N.; Colonel T. Eustace Smith, C.B.E., T.D., D.L., President, The Shipbuilding Conference; E. L. Denny, Esq., B.Sc., Past President; Dr. G. B. B. M. Sutherland, F.R.S., Director, The National Physical Laboratory; Captain W. S. Dawson, B.S., S.M., U.S.N., representing the United States Naval Attaché; Captain F. N. F. Johnston, D.S.C., R.N.Z.N.; R. B. Shepheard, Esq., C.B.E., B.Sc., Director, The Shipbuilding Conference; Captain R. S. David, I.N.; The Reverend Maurice Dean, B.A., The Rector, St. Olave's, Hart Street, London, E.C.3.; Dr. T. W. F. Brown, C.B.E., S.M., Research Director, Pametrada; R. Cook, Esq., M.Sc., Chairman of Council; D. C. Haselgrove, Esq., Under Secretary, Ministry of Transport; B. C. Curling, Esq., O.B.E., Honorary Member; Captain R. Gabbett-Mulhallen, Captain Superintendent, H.M.S. *Worcester*; Victor Wilkins, Esq., F.R.I.B.A.; Commander M. A. K. Lodi, P.N.; H. Desmond Carter, Esq., President, The Institution of Mechanical Engineers; A. W. Wood, Esq., Assistant Secretary, Ministry of Transport; E. C. V. Goad, Esq., Assistant Secretary, Ministry of Transport; E. J. Hunter, Esq., B.Sc., President, The North East Coast Institution of Engineers and Shipbuilders; R. H. E. Thomas, Esq., O.B.E., Vice-President, The Institute of Fuel; R. W. Bullmore, Esq., M.B.E., Director of Sea Transport, Ministry of Transport; D. S. Tennant, Esq., C.B.E., Secretary, The Merchant Navy and Air Line Officers Association; R. W. Reynolds-Davies, Esq., O.B.E., B.Sc., Secretary, The Institute of Fuel; B. G. Robbins, Esq., M.Sc., Secretary, The Institution of Mechanical Engineers; D. S. D. Williams, Esq., Past President, The Diesel Engineers and Users Association; T. S. Nicol, Esq., Secretary, The North East Coast Institution of Engineers and Shipbuilders; H. F. Hesketh, Esq., President, The Society of Consulting Marine Engineers and Ship Surveyors; J. D. C. Stone, Esq., F.C.A.; Captain A. D. Duckworth, R.N., Secretary, The Royal Institution of Naval Architects; Ronald Ward, Esq., F.R.I.B.A.; A. P. Quarrell, Esq., Secretary, The Diesel Engineers and Users Association; H. A. J. Silley, Esq., C.B.E., Past President.

The Loyal Toasts, proposed by the Chairman, having been honoured, His Excellency Baron Adolph Bentinck (The Netherlands Ambassador) proposed the toast of "The Royal and Merchant Navies of the British Commonwealth".

He said: I am very honoured to be here today and to address you. It is a formidable task, and I am all the more grateful to you, Mr. President, that you gave me the chance to accumulate some Dutch courage beforehand! (*Laughter.*) I know that I have been requested to speak to you tonight because I am the representative in London of a generally friendly seafaring nation.

All the same, when I received the request to speak to you, I was rather puzzled by the text of the toast I was asked to propose tonight, "The Royal and Merchant Navies of the British Commonwealth". I have not been very long in London—only a year and a half—but I understand that the Royal Navies are what one would call the fighting Navies; and as far as my knowledge of the Commonwealth goes, there are two countries which belong to the Commonwealth and whose fighting Navies no longer have the prefix "Royal". I am thinking of the Indian and Pakistani Navies. Nevertheless, we Dutch know from experience that a Commonwealth Navy, even though it is not "royal", can be a very formidable opponent, as we learned in the days of Oliver Cromwell. (*Laughter.*) I have since been informed that the title of the toast I am to propose is the traditional one, and I am sure that it includes all the fighting Navies of the Commonwealth.

If I may revert to my own country, our and your fighting Navies and our Merchant Navy have had their differences in the past. It is certainly not my job to enlarge on these differences. On the contrary, I should like to remind you that we have often stood shoulder to shoulder and that our two Navies have fought common battles. I would remind you of the Armada. We both, your Navy and ours, captured Gibraltar, but you kept it. Whether we would like to change places with you at this moment, I am not quite sure.

One common struggle which I should certainly like to remember tonight is the last war. Following the invasion of our country, our Navy lost its home, but it found a home from home in these Islands. With the support not only of your Navy under the White Ensign, but of all your Services, it was possible for the Netherlands Navy to play its part in the allied battles on all the seven seas—the Mediterranean, the North Sea, the Atlantic, the Pacific, the Java Sea and everywhere sea battles were fought. We are most grateful to you for the help you gave us, and I hope you feel that we have played our part in a satisfactory way. (*Hear, hear.*) In those difficult days, it was certainly the fighting Navies of the Commonwealth who, with your and our Navies, played a big part to secure victory for the cause of liberty and freedom.

Since the war, our Royal Navies have co-operated very closely in the North Atlantic Treaty Organization. The difference this time, however, is that we co-operate not to win battles, but to fulfil the aim of the North Atlantic Treaty to avoid the necessity of fighting battles. We co-operate in order to preserve the peace. (*Applause.*)

According to historians, the Dutch ships at one time went to sea with a broom in the mast to clean the seas for our trade. The British were far more subtle and, I would say, reaped the same or even greater advantages, not by sending their ships to sea with a broom, but with a duster—the "Red Duster".

It is my pleasure today to pay tribute to the Merchant Navies of your country and of the Commonwealth as a whole. They perform a great service for the benefit of mankind in carrying the world's goods from country to country in the most efficient and best ways.

Here again, in the question of the Merchant Navy and what it does, your country and mine stand shoulder to shoulder in our opposition to flag discrimination. Your country as well as mine indulged in this practice in days gone by, but certainly the practice of flag discrimination is out of date in these modern times, when we all strive to improve the wellbeing of mankind. All measures that tend to eliminate free competition, which is the best way of improving the wellbeing of mankind, *must* be avoided. (*Hear, hear.*)

The Merchant Navies do not only carry goods: they also carry passengers. That reminds me of the story of a man who was on board a passenger liner—not one that was fitted with your Chairman's stabilizers. His wife said to him, "Look, John, over there. What an enormous ship". The husband, who was rather seasick, replied, "I do not want to see any ships. Call me when you see a bus". (*Laughter.*) I am sure that this husband was a Londoner. It is very curious that in the big cities like London, the buses tend to adopt certain naval characteristics; for example, as you must have noticed, they tend to travel in convoys! (*Laughter.*)

There is no doubt that through the centuries things at sea have changed. This was quite clear to the young middy who was received by his captain for the first time. The captain said to the young man, "Well, boy, the old story, I suppose—the fool of the family sent to sea". "Oh, no, sir", replied the middy, "that is all changed since your days". (*Laughter.*)

I am glad to say that one thing has not changed, and that is the spirit of co-operation and comradeship between all seafarers and between the fighting and merchant Navies of our countries. This co-operation, especially between the merchant and the fighting Navies, is best symbolized by a flag of which I have not yet spoken—the Blue Ensign. It is a great privilege for the ships which are allowed to carry that flag, because their masters have qualified for the Royal Naval Reserve and they symbolize the essential co-operation between those two sides of our shipping business and of our fleets.

With that spirit in mind, I am especially happy and honoured tonight to propose the toast of "The Royal and Merchant Navies of the British Commonwealth". (*Applause.*)

The Right Honourable Ernest Marples, M.P. responded, and proposed the toast of "The Institute of Marine Engineers".

He said: I would like to start by saying that we have listened to a brilliant, astonishing and versatile speech from a man whose native language is not English. (*Hear, hear.*) It was an incredible performance of humour, sense and comradeship which I shall remember, particularly as I had a Dutch girl friend in my early days. (*Laughter.*) I was persuaded to make a speech in French in Paris at the instigation of my Dutch girl and everybody present realized in about two minutes that my French had been learnt in the most agreeable but the less respectable parts of Paris. (*Laughter.*) We are grateful to you, sir, for all you have said. We have noted it and we thank you for your kind good wishes.

On the second Friday in March three years ago, my predecessor, Harold Watkinson, did two things. He inaugurated work on the Inner Ring road at Birmingham and then rushed back to speak to the Institute of Marine Engineers in the evening. Today is the second Friday in March three years later. I opened the first section of the Inner Ring road at Birmingham today and rushed back to dine with the Institute of Marine Engineers in the evening.

You may think that three years is a long time for an inner ring road to be completed, but I would remind you that during the last war, to which His Excellency has referred, the Americans gave millions of dollars for the production of aircraft, and they put in charge of the production of aircraft, for some unknown reason, an admiral. He was criticized mercilessly by the Press because there were no aeroplanes and a lot of dollars were flowing out. Like all admirals, he was livid and annoyed. He opened a hospital in New York and thought that this was his chance to make his mark. So he said, "Gentlemen, this hospital is the finest hospital in the world. It has the best physicians, the best X-ray apparatus and the best gynaecologists, but, gentlemen, it still takes nine months". (*Laughter.*)

That illustrates my point that, as a Minister, all I can do is to repeat the quotation, "Thought is easy; action is difficult". To act in accordance with one's thoughts is the most difficult thing in the world, except for one class of person—the man who sits in the editorial chair of a daily newspaper. He starts at 7 in the evening with a cigar and finishes at 7.45, having told the nation the solution to all its problems, and then goes home. A fortnight later he says: "Why haven't the Government done something?" It has been the epitome of my ambition, throughout the vicissitudes of my life, to end in the editorial chair. What I should have to say would be nobody's business!

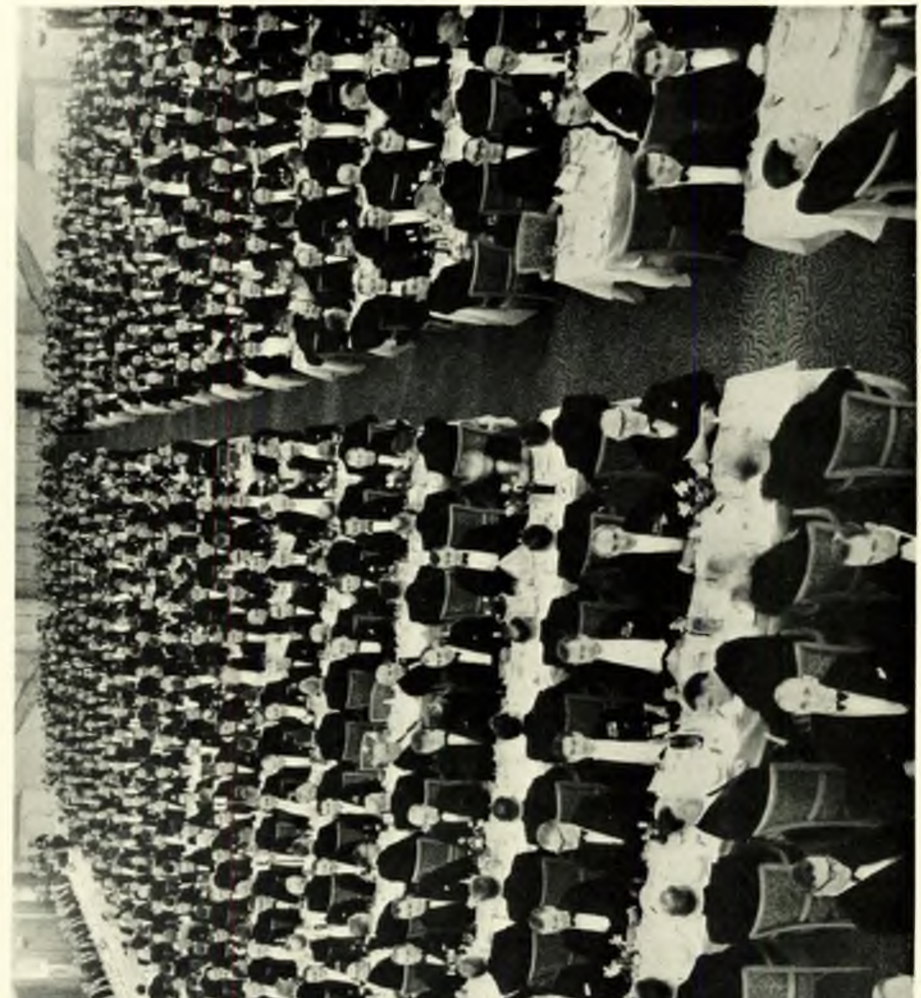
This is the first time that you have had a Minister of Transport speak to you in his capacity of being responsible for shipping, shipbuilding and ship repairing, in addition to roads and something I have heard about lately—railways. When I went to No. 10, Downing Street after the General Election, Mr. Macmillan said to me, "I would like you to take on transport. There is the problem of the roads—very difficult; many motor cars and few roads. There is rail, even more difficult—bankrupt. Then there is shipping, facing a depression never known before". I said, "Thank you. I would be even more grateful if you would indicate to me what I should do in my spare time". (*Laughter.*)

In private life, I used to be happy as a civil engineering contractor. We then had in civil engineering similar problems to you now in the marine world. The engineers were too few, and too few apprentices were coming in, as is happening today. It is difficult, and there are two reasons for that. The first, I think, is full employment, because in a democracy where there is full employment people tend to choose the easy form of employment instead of the hard and less rewarding one. They want a job ashore which is permanent and relatively easy. They marry early, which is a good thing in some ways. A woman accepts a man for the sake of matrimony; a man accepts matrimony for the sake of the woman. I think Kipling was quite right when he described Captain Gadswick, of the Pink Hussars—not the Pink Zone: "He made a good husband and thereby became a bad officer". It may be true of our engineers.

But now I have a brief from my Department which I must read to you. It says that a generation or more ago, there were frequently many certificated engineers serving in the average ship, and now many companies count themselves lucky if they have more than the minimum required to comply with the legislative requirements as regards manning.

I believe, frankly, that the question of apprentices and engineers and technicians will be crucial to our future health in all walks of life—(*Hear, hear*)—civil, marine, electrical and mechanical. We must produce the engineers. We must try, if we can, to give them the desire to make themselves efficient, to rise in the world, and to give them the rewards which are their due if they so rise.

I should like to talk about nuclear power. When I listened to His Excellency the Ambassador, who made such an excellent speech in English, I was reminded of when I flew to Berlin in an aircraft during the airlift. The week afterwards, I was about to make my first lecture tour in America, and so I thought I would practise on the officers and the flight-sergeant in the aircraft; they were all American. For twenty minutes I explained what life was like in Berlin to an American sergeant from Alabama. He listened carefully and intently and asked a series of questions. I am ashamed to say that when we arrived at Frankfurt, he hit me between the shoulder blades a very severe blow, much too vigorously, and said, "Say, Bud, I wish to congratulate you". With that reticence which we all



Annual Dinner, 1960

Sir William Wallace, C.B.E., LL.D., The Right Honourable Ernest Marples, M.P., and Mr. H. Desmond Carter

The Right Honourable The Lord Winster, P.C., K.C.M.G., His Excellency Senor Victor Santa Cruz, and Sir Gilmour Jenkins, K.C.B., K.B.E., M.C.

have, I was delighted to have congratulations. He then added, "For a German, you speak good English". (*Laughter.*)

When I went to the United States of America recently, I discussed road and rail problems very vigorously, and shipping even more vigorously; because although the Dutch Ambassador in Washington is the spokesman for all the European shipping nations, as I was passing through Washington I felt that as a Minister I should not miss the opportunity of seeing the State Department of America and letting them know my personal views on shipping, discrimination, subsidies and flag discrimination—which I did. I said some nasty things, but some nice things as well, because I believe that we must have freedom in shipping if we in Europe are to play our part.

Frankly, and I have said this in the House of Commons, I cannot reconcile the protestations of our American friends—and, indeed, they are friends and allies—about freeing trade and having fewer restrictions on dollars and other things with their practice in shipping. I am bound to say that I do not like the way that certain ships are subsidized and 50 per cent has to be carried in certain ships. No country in the world can stand competition of that nature. I made my view known quite firmly and clearly.

On the other hand, of course, they have a shipping lobby You know what democracy is. Bernard Shaw said that it was the election by the incompetent many instead of the appointment by the corrupt few. Sir Winston Churchill once said that it was the worst form of Government ever known, with the exception of all the other forms that had ever been tried.

I must say, however, that the Americans were most helpful in showing me their new ship, the *Savannah,* and all that goes with it. They have had our experts there and I am convinced that the United Kingdom must play its full part in the development of this new technique. I am quite certain of that. Unfortunately, however, technical developments are going fast; they are probably ready to go into production. As you know, in the Ministry of Transport we have tenders out for two types of nuclear propelling machinery. It is quite certain that the first ship will not be economic, but we must go on trying until we get economy. We have had a committee on safety, in which many of you have played your part. I am grateful to you. It was announced in the House of Commons that we have had a comprehensive and adequate report, on which we will act.

In this country, too, there is a misunderstanding of the Americans, just as in America there is a misunderstanding of the European. I was once in America when there was an all-in wrestling match with an Englishman called Gorgeous George, who had fair, wavy hair, and a second called Lord Fauntleroy, in a dinner jacket, who squirted scent on him as he went to the middle of the ring. It is just as bad in this country, because I remember what happened during the war in Wiltshire. The lady who was in charge of the village had lived in her house for 150 years, and the American Eighth Air Force came to stay and were billeted in the grounds. They had a jeep per man—everything, in fact. At the end of a week, the Vicar went to her and said he was very sorry but hoped she had not been unduly disturbed. "Not a bit", said the lady, "I like the Americans. They are perfectly courteous, civilized and generous—with the exception," she added, "of those peculiar white people they brought with them". (*Laughter.*)

I am sure that we in this country must always remember three things, three things which I personally find stimulating, in spite of my many difficulties. First, the thing that characterizes the Western European more than anything is his spirit. If you give a man the right spirit, there is no knowing what he can do. It is the spirit that counts, not the letter of a contract. One has only to look at the marriage contract, short though it is, to see how differently it is interpreted! Buy the *News of the World* this week and read the principal article in it, by Diana Dors. (*Laughter.*) Some months ago, she said she was going to settle down, farm and raise a family. The whole essence of farming is perpetual pregnancy.

The second thing is to let us have our character. Character is fashioned by adversity. We face problems on road and rail, and in our balance of payments—but why not? If we did not have problems, we should not be human beings. A human being goes to pieces under conditions of ease. It is a problem that is challenging and stimulating.

Thirdly, let us have ability in our schools. Let us seek not only to learn, but to think, and seek not only to accept, but to question as well.

If we have these things at the back of our minds, we—the Dutch, the English, the French and all the Western Europeans—will be able to contribute to history in the future the same great things that we have contributed in the past.

Sir, I am grateful to you for asking me here. If you think at any time that in my capacity as Minister of Transport I can assist, I hope that you will not hesitate to approach me. I am grateful to you for listening to me so patiently and so kindly.

I now have the pleasure of proposing the toast of "The Institute of Marine Engineers". (*Applause.*)

The PRESIDENT, in response, said: I must first crave your sympathy. We have had a representative from Holland speaking perfect English and bringing roars of laughter and cheers, followed by the Minister, who again departs from his serious ideas and tries to cheer us up. He cheered us up so much that he very nearly forgot to say, "You are decent fellows and I am going to propose your health". (*Laughter.*)

I thank you, sir, for honouring us with your presence tonight. We realized that in the public eye you, as Minister of Transport, have probably the toughest job. You have a Ministry which affects all members of the community and, as such, produces all the criticism in the world. We are indeed grateful to you, sir, for giving us your time.

We are grateful to His Excellency the Netherlands Ambassador for coming here and proposing so ably that awkward toast which I dare not repeat and which I should get mixed up. I should probably get the Commonwealth mixed with the Navy and the Navy with the Army. We are also grateful to the Minister of Transport for proposing the toast of "The Institute of Marine Engineers".

I was very grateful when the Minister from another great nation—the Chilean Ambassador—said: "Where do you get, in any country, 1,200 funny-looking chaps, all technically able, in one room covering one profession?" I said that many of them were pretty good and a few pretty dud, but that we had also brought in a few naval architects. (*Laughter.*) I got my orders from the Secretary, whereas the others who have spoken had a free hand, but I really must say "Thank you" for coming.

I also desire to welcome, on behalf of the Institute, Their Excellencies, and to hear the gracious Ambassador whose name I thought I could never remember but I am perfectly sure it is Baron Adolph Bentinck, not forgetting that on my right, as the boxers say, we have the Chilean Ambassador, Senor Victor Santa Cruz. (*Applause.*)

We are honoured with the presence of the First Lord of the Admiralty. Thank you, my Lord, for coming along to see us. (*Applause.*)

We have, of course, also our Past Presidents, who have come to see what sort of mess their successor will make of this dinner. That has been proved already.

Also, we have representatives of the Commonwealth Navies; and I really must mention the Privy Council, because there are certain things we like them to do for us and so we must remember their presence here tonight.

We have also with us representatives from practically all our kindred associations. Among the important people I should have mentioned Lloyd's Register, because, heaven knows, they give us trouble in plenty in their endeavour to keep us operating on sound lines.

I particularly wish to mention tonight our kindred association, the Royal Institution of Naval Architects. This is our first opportunity to recognize publicly the swanking institute, the "Royal" Institution.

We are making progress. Clearly, my predecessors and the Chairmen of our Council had courage in looking so far forward. The Memorial Building, which we set up and built to honour the memory of all the marine engineers who lost their lives in the struggle for freedom, is not only a monument, but is a monument to the courage of my predecessors who faced very difficult problems, financial and otherwise. (*Applause.*)

Like Sir Ronald Garrett and many of my predecessors, I have been privileged to visit many of what are really the pillars of our Institute—that is, the Sections. These Sections are flourishing in all parts of the world. As one finds Scotsmen abroad celebrating St. Andrew's Day and Burns' night as they never do it at home, so I think one finds when visiting the Sections abroad that they are full of enthusiasm. I know that when, as President-elect, I was allowed, having been kicked out of my company, to amble round the world for a little while, I found great enthusiasm and was very well fed and welcomed at any of the Sections I visited.

As a Scot, I am really a little worried, because in Scotland we have a really good Section. But, after all, according to tradition, Scotland is the home of the marine engineer. I am going to urge them to do a darned sight better, because when I went to Birmingham, which is about as far away from the sea as one can get in this little island of ours, I found an enthusiastic Section of 250 members, all darned good fellows, with better jobs than ever they had at sea! (*Laughter.*)

Mr. Minister, you have mentioned America and we are grateful to you, because we know that in Washington you spent many hours pointing out the absolute necessity of freedom for shipping if Britain is to maintain its position. We are very grateful to you, sir, with all your worries, that you have done that for us. (*Hear, hear.*)

I would like, on behalf of all here, to make a plea to another Ministry to help us to gain freedom in endeavour. We are now, as you all know, in competition as shipbuilders and engineers with the whole world, but we are continuously frustrated in our efforts to modernize and bring our production up to date by our trade unions. It is appalling that the Government, having sent many abroad to see what is being done, they still stick to the horrible habits of the past. We are continuously up against the awful problem of demarcation. The Minister and many here have been about abroad, and any number here could quote instances of a simple production job in this country calling for six tradesmen when it could have been done efficiently by one tradesman. And so I make this appeal to another Ministry to do all they can to help us. (*Applause.*)

I am now thinking of the prestige of our Institute, which on this occasion made a blunder and elected one of its members as President—a very stupid thing to do; but I am proud tonight that we have with us our President-elect, Lord Simon. Lord Simon will take over in April. I am sure that all those who heard his inspiring address to us some few years ago will agree that we can feel proud and congratulate ourselves. I am sure that our Chairman, our Council and our members are looking forward eagerly, sir, to your term of office. That Lord Simon, with the extremely responsible position which he holds today, has agreed to serve us means that somebody must be congratulated upon persuading him to do it.

I also have a brief. I must tell you about the good work our fellows are doing, otherwise I shall be strafed by the Secretary. In any case, I am sure that all who are here would like me to move a really sincere vote of thanks to our Chairman and our Council and our Section Committees for the vast amount of work they have done, and the energy they have put into it, in the interests of our Institute. (*Applause.*)

I have attended meetings of the Council and it is tremendously encouraging to see senior members of the Navy, the Merchant Service and our shipowning community who are prepared to give up their time to this, at times, rather dull job. They at least have had recognition from other bodies, as almost without exception we now have representatives on many technical colleges and technical organizations.

We have been trying to do something abroad, and I should like on this occasion to say how proud we are of the outcome of the visit of our Chairman of Council and the Secretary to America and Canada, which resulted in the very cordial meeting which was held in New York, a combined meeting of our people with the body representing the Marine Engineers and the Naval Architects of New York. (*Applause.*)

I was pleased recently to see that Sir Nicholas Cayzer, with his great knowledge, is pressing hard for the scrapping of old ships and the building of new ones. I can assure you that we marine engineers are gasping to see new ships at sea with modern machinery manned enthusiastically by our members, despite the fact that the poor "chief" will be very worried with the more and more complicated equipment which the backroom boys are forcing upon him. Lord Geddes recently reminded us that we are all out for 95 per cent efficiency and suggested that we might think again as the efficiency that we were aiming for would cost the shipowners far too much money; and so he asked us to go easy.

Anyhow, I suppose, we shall soon be carrying electronic engineers. I see that one of our latest liners is to be equipped with the most elaborate television in the world. What the poor "chief" will do, I do not know, unless he gets specialists on his staff. The other day, I heard of a man who had been transferred suddenly from oil burning steam reciprocating to one of those awful free piston gas turbine jobs and was expected to do it efficiently.

However, judging by the papers—now I am advertising—that are read and discussed at our meetings, and the amount of information that is available to our members, I feel sure that our members will tackle all the designs that the back room boys are prepared to put up. If the shipowners, with the advice or otherwise of their technical people, are prepared to install them in their ships, they can do so, confident that you will all go ahead and maintain them efficiently.

The Institute is anxious to build up membership higher and higher, and so we are going to ask the superintendents to do all they can to help us in this matter.

Again, sir, on behalf of the Institute, we are most grateful to you for coming here. (*Applause.*)

The proceedings then terminated.

INSTITUTE ACTIVITIES

Section Meetings

Bombay

General Meeting

At a meeting of the Bombay Section held on 27th February 1960 at the Naval Engineering College, I.N.S. *Shivaji,* at Lonavla, Mr. N. J. D'Sylva re-presented his paper on "Selection of Lubricants for Marine Machinery" for the benefit of members resident there and the staff and students of the College. The meeting was well attended.

Joint Meeting

A joint meeting with the Institution of Marine Technologists and the Company of Master Mariners of India was held on 25th March 1960 at Scindia House, Ballard Estate, Bombay, when Professor E. V. Telfer, D.Sc., Ph.D., gave lectures on "Striking Ships" and "Ship Service Analysis". Rear Admiral T. B. Bose, I.N. (Local Vice-President) presided. Several members asked questions to which the speaker replied.

Captain S. B. Aga, General Secretary of the Company of Master Mariners of India, proposed a vote of thanks to the Chair and the speaker and to the Scindia Steam Navigation Co. Ltd. for the use of their hall. The meeting ended at 8.0 p.m.

Annual Boat Cruise

The Annual Boat Cruise was held on 24th March 1960 at 7.30 p.m. The m.v. *Shobana* was made available to the Section for the cruise by Scindia Workshop (Private) Ltd., through Mr. D. B. Daruwala (Member of Committee). Members with their wives and guest totalling 126 attended. A buffet dinner was provided on board and entertainment was provided by the orchestra of the Dockyard Apprentice School Boys, through the courtesy of Commander W. P. Bapat (Member of Committee).

Rear Admiral T. B. Bose (Local Vice-President) thanked Scindia Workshops, Mr. Daruwala and Commander Bapat, and the members of the subcommittee responsible for the function (Messrs. B. S. Sood, R. G. Sathaye, D. B. Daruwala, D. Dyer and C. S. Sundaram), who had all assisted in making the occasion a success. The cruise ended at 10.30 p.m.

Scottish

On the 16th March 1960 a visit was made to the Valleyfield Paper Mills of Alexander Cowan and Sons Ltd., Penicuik, when a paper entitled "Modern Papermaking" was read by J. A. Walker.

Prior to the reading of the paper members and visitors, numbering about fifty from the Glasgow and Edinburgh areas, made a tour of the mills and were shown all the various stages of papermaking, which proved most interesting. When the tour ended, tea was served through the courtesy of their hosts, and the meeting terminated at 6.15 p.m.

West of England

A general meeting of the West of England Section was held at Smith's Assembly Rooms, Bath, on Monday, 14th March 1960 at 7.30 p.m. In the unavoidable absence of the Chairman, Mr. F. C. Tottle, M.B.E. (Vice-Chairman of the Section) presided, and there was an audience of thirty, including the Local Vice-President, Mr. D. W. Gelling.

The Vice-Chairman then introduced Mr. S. Archer, M.Sc. (Member) who gave a very interesting lecture entitled "Some Notes on Recent Reduction Gears for Propulsion Purposes", which he illustrated with numerous lantern slides.

Eight speakers took part in the discussion that followed, and Mr. Archer dealt with the many questions in a masterly and agreeable manner.

On the proposal of Mr. J. B. Goodier (Associate), Mr. Archer was accorded an enthusiastic vote of thanks for his excellent lecture.

In closing the meeting at 9.20 p.m. the Vice-Chairman said that if any members would care to remain behind, Mr. Archer would be pleased to show a few additional slides illustrating some defects found in gear teeth. Such was the interest shown by members that all present remained seated until the last slide had been shown at 9.40 p.m.

Corrigendum

The information given in the last seven lines of the annual report of the Southern Joint Branch R.I.N.A. and I.Mar.E., which appeared on page xxiii of the March 1960 Transactions, was incorrect in certain respects, and should have read:

> "At the Annual Dinner, which was held at the Royal Beach Hotel, Southsea, on 30th October, in addition to officials of the parent societies, they were honoured by the company of Captain G. Villar, C.B.E., R.N. (ret.) as guest of honour, and by Mr. I. K. King, C.B.E., Director of Dockyard Division, Admiralty, and Mr. J. H. B. Chapman, C.B., Director of Naval Construction, as private guests".

Election of Members

Elected on 14th March 1960

MEMBERS

Edward Barber
Donald James Berry
John Davenport Buckley, Lieut. Cdr., R.N.
Adam Drennan
Stanley Walton Edward
Jack Armstrong Fenn Clark, Cdr., R.N.
Robert Riviere Hunter
John George King
Leif Erling Korner
Bruce Durward Lawson
George McRobbie McCallum
Duncan Charles Reid Macfarquhar
Alfred Arthur McGlashon
Ferdinand Friedrich Mattern
John Hugh Matthews
John Bell Melville
Robert MacLean Paterson
Edmund Hort Player, Cdr., D.S.C., R.N.
Leslie Cowan Rennie
Jack Templeton Robert
Eric Durose Scragg
Frank Soutar
John Clark Stevenson
Anthony Leonard Thackara, Cdr., R.N.
Phillip George Thomas
Arthur Campbell Waldie
John Williams

ASSOCIATE MEMBERS

Wilfrid Collinge Ainley

Edward Barlow
R. D. Bhave
Geoffrey Paul Brown
Rex Calton
Brian Peter Scott Charlton
Leonard Henry Christopher
John Henry Collins
Allan Francis Crooks
Andrew Cushnaghan
Herbert Denzil Dunn
John Edwyn Ferrier, Lieut., R.A.N.
Ernest Norval Geldart
Daniel Gilbert, M.Sc.(Manch.)
Robert McGonn Green
Glen David Havard
Malcolm James Henderson
Raymond Reginald Hooker
Douglas Hughes
Walter Henry Jacques
Eric Ramsay Kendall
Victor M. Kennedy
Arthur William Laws
John Angus Lazaras
William Brown Leitch
Jeremiah McDonald
Philip John Maguire
Harold John Miller, B.Sc.(Durham)
William Leslie Morrison
M. O. Oommen, Lieut., I.N.
Napier Herbert Kitchener Page
James Penny
Harry Frank Pitcher, B.Sc.(Eng.) London
William Porter
James Rodger
Bernard Rose
Maximilian Egon Rosner, B.Sc.(Eng.) London
Robert John Malcolm Rowe
Henry Robert Selby
Ronald Alexander Smyth
Alexander Leslie Wason Stevens
Joseph Trevor Swaine
David Anderson Turner
James Watson
Derek Williams

ASSOCIATES

James Baillie
James Robert Barlow
Reginald Edward Bowman
Ralph Hugh Friedlander, Captain
Peter William Jacobs
Francisco Lopez Carlos-de-Vergara
Hugh McNallen
Henry Forbes Mitchell
Alec Geoffrey Scott
Ronald Thomas Taylor
John Patrick Twohig
Louis Bertie Whalley

GRADUATES

Stewart Adamson
Roy Walter Bryant
David John Cowley
Malcolm Quentin Dickson
James Macdonald Earsman
Gerald Taylor Gailey
Ian Douglas McGilp
Alexander Campbell Murdoch
Maung Aung Pe
Douglas Albert Richard Rosario
Gerald Henry Rundle
John Pow Taylor
Gerald Arthur Stephen Wilkes

STUDENTS

Ronaq Raza Abidi
William Joseph Brickley
Michael Bruce
Martin John Challis
Neil Mackinnon Churcher
Neil Raymond Dalton
Roger Burnett Davis
Robert Edward Davy
Oskar Ernst Evensen
Peter Michael Farrell
Howard Paul Foreman
William Grant
John Henry Hansen
Gordon Langley
John Liddane
Samuel Oyerinde Oyediran
Nicolas Stamatakis
Christopher John Triffit
David Wanless
Eric Wynne Davies
Robert Leslie Hook
Donald Compton Wood

PROBATIONER STUDENTS

David Charles Ashton
P. H. Burdon
Roderick Barry Cotton
Edward Alexander Cuthill
Robert Alexander Donaldson
Michael John Earp
David Frederick Fletcher
Anthony Grover
David John Gurney
Anthony Alexander Henley
Brian G. Hooper
John Demaid Jones
Richard Oliver
Eric John Poolton
Stephen Frank Stallwood
Knock Hing Tham

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER

Arthur Noel Stuart Burnett, Lieut. Cdr., R.N.
Eric George Hickling
Paul Darneley Hobson
Artur J. W. de Pessoa Lobo
Donald Campbell Nicolson
Charles Gerard Purvis
Anthony Claude Soward

TRANSFER FROM ASSOCIATE TO MEMBER

Herbert James Aspin
Leslie Gordon Duncan
Leslie Greenacre
Alexander Foster Harrold
Murray Robert Osborne
Henry Alexander Sledge
Joseph Symes
Arthur Hermon White

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

P. A. D. Anbu, Lieut.(E), I.N.
Ernest George Knight

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Peter William Chandler
Henry Arthur Curnow
Alan Greenhalgh
Thomas Parker
Norman Schofield
George Simpson
Thomas Jeffrey Stedman
James Strachan
Brooking Young

TRANSFER FROM PROBATIONER STUDENT TO GRADUATE
George Patrick Brown
Keith Francis Jones
Alan Digby Makinson
John Edward Milham

TRANSFER FROM PROBATIONER STUDENT TO STUDENT
Ian Jeffrey Day
Patrick Williamson Lee
Thomas Anthony Machell
Frederick Joseph Parle
Roger James Parry
David Edward Pelton
Ronald Mervyn Pereira
Anthony John Pulford
John Michael Reeve
Jan Alexander Taylor
Richard James Taylor
Derek Totton
Kenneth Gordon Wheatley

Elected on 11th April 1960

MEMBERS
George Ernest Adams
Joaquim de Carvalho Afonso
Harold Bates
Arthur Peter Beard
Leslie Arthur Broomfield
George Alfred Brown
E. Conolly
Richard Evans
James Charles Gabriel
Robert George Alexander Lawrence
Kenneth Ewart Lewis, Cdr., R.C.N.
Erik Mathiesen
Kuldip Singh Oberoi
Geoffrey James Parker
William Abraham Rae
Robert Charles H. Reed, Cdr., D.S.C., R.D., R.C.N.R.(R)
Norman Cecil Rivett
Joseph Schormann
Lawrence Leonard Smaill
John Richard Stott, B.Sc.(London), B.A.(Cantab.)
Reginald Warren

COMPANIONS
John Harrison Houston Jackson
Gordon Wilson Stead

ASSOCIATE MEMBERS
Colin Jocelyn Andrew
Arthur Waller Bownass
Dennis Anthony Clarke
Brian Joseph Crangle, B.Sc.
Robert Hall Curry, B.Sc.(Durham)
John David Egginton, B.A.(Cantab.)
Bryan William Evans
Anthony Norman Haigh, Lieut. R.N.
Malcolm Hart
Peter Johnston
Stephen Joseph, Lieut., I.N.
Russy Manekshaw Kalyaniwalla
Arthur King
William Gilbert Marsh
Dr. Ing. Giovanni Mazzini
Peter Ramsay Mitchell
David Wilson Neish
John Owen
Robert Patterson
Douglas Frederick Porter
James Henry Pull
Raghavan Varada Rajan
Ernest Arthur Roberts, B.A.(Cantab.)
Terence Antony Dyson Sharp
Ronald Taylor
Thomas Morton Veitch
James Denis Wilkinson

ASSOCIATES
Reginald Roy George Best
Robert Charles Brown
John Hay Davidson
William Gordon
Olivier Jean
David William Llewellyn Jones
Harold Unwin

GRADUATES
Peter William Chapman
Robert Bayne Truscott Donald
Basil Leonard Gridley
David McGlashan
Ian Malcolm Moffat
John Ramsay Murphy
James Mair Murray
Thiagalingam Nirmalalingam
Edward James Sedgewick
Robert Edward Sherman
Kenneth Sinclair

STUDENTS
Clive Edward Brown
Bruce William Crewes
Michael Edwin Gordon Hadlow
Badrul Islam
Reginald Keating
James Leslie Kelly
Kenneth John Maynard
Michael John Perkins
Michael Woodward

PROBATIONER STUDENTS
Philip Melvyn Bradbury
George Islay Gardner
Thomas Hume, Jnr.

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER
Harold Clark
Arthur Greener
Edward Ronald Jones
Arthur Charles Parkinson
Robert Colquhoun Stewart
Robert Yarr

TRANSFER FROM ASSOCIATE TO MEMBER
Robert Bowie
John Mervyn Mackenzie

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER
Clarence Athol Bell
Helmy Ahmed Helmy

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER
Ramesh Chandra Bansal
Alexander Amos Jack Couchman, B.Sc.
Ajoy Dutta
William Keith Highfield
Lakshmi Narayan Misra
Roy Maurice Spencer
Ronald Whitaker

TRANSFER FROM PROBATIONER STUDENT TO GRADUATE
Brian Edward Bowes
Percy George Munn

TRANSFER FROM PROBATIONER STUDENT TO STUDENT
John Cooper
Ian Michael Huckle
Trevor George Robinson
Kieran Justin Shanahan
Michael Bulkeley Smalley

OBITUARY

FRANKLIN THOMAS ANDREWS (Member 7640), former commodore chief engineer officer in the British India Steam Navigation Co. Ltd., died at his home in Southampton on 17th January 1960. He was fifty-nine years old.

Mr. Andrews joined the British India Line as a junior engineer officer in July 1924, his first appointment being that famous pre-war trooper *Neuralia*. In May 1927 he was promoted fourth engineer officer of *Angora*, subsequently gaining promotion to third engineer officer some eight months later.

The second world war saw Mr. Andrews as second engineer officer of another B.I. trooper, *Dilwara*, having joined the vessel when she was building in 1936. He remained in this ship for the greater part of the war, until his promotion to chief engineer officer in September 1944. However, he was destined to return to *Dilwara* for the remaining years of his service with the B.I., and after having served on various vessels, including the cadetship *Chantala*, he returned to the troopship in May 1956. He was appointed commodore chief engineer officer on 29th October 1957 and retired from the sea on 11th July 1958.

Mr. Andrews joined the Institute as a Member in 1934.

REGINALD HENRY CHANTER (Member 4494) died, aged sixty-two, on 28th March 1960. He was apprenticed to the General Steam Navigation Co. Ltd., Deptford, in 1913 and went to sea in ships of the company from 1916. He joined the Institute as an Associate Member in 1922 when he obtained a First Class Board of Trade Certificate. Mr. Chanter first sailed as chief engineer in 1924 and remained in that capacity with the General Steam Navigation Company throughout the second world war, resigning in 1946.

For the next ten years Mr. Chanter was chief maintenance engineer with Goodlass Wall and Co. Ltd., Millwall Works, and from 1955 until his death he was chief engineer of a family concern, the Crusader Manufacturing Co. Ltd., Walthamstow.

Mr. Chanter had a long association with the Merchant Navy and Air Line Officers' Association, having been a member of their Representative Council (of the Officers' (Merchant Navy) Federation) from 1933/42 and a member of the Executive Council from 1934/46.

JOHN COX DANIELS (Member 7021) died on 15th November 1959, aged fifty-seven.

He joined Canadian Pacific Railways as an apprentice engineer at their Bootle Engine Works in 1918 and in 1923 was appointed seagoing assistant engineer in the old *Empress of Britain*. He obtained a First Class Board of Trade Steam Certificate in 1932 and then served for six years in the *Duchess of York* in positions up to fourth engineer. In 1940 he was in the new *Empress of Britain* when she was sunk; and two years later he was in the *Winnipeg II* when she was sunk. Mr. Daniels was appointed staff chief engineer of the *Empress of Canada* in 1951 and served in her until she was destroyed by fire in dock in 1953. He was afterwards appointed chief engineer of the *Beaverford* and served in her until his retirement through ill health in June 1956.

Mr. Daniels had been a Member of the Institute since 1932.

WILLIAM JOHN JOYNSON (Member 10134) was apprenticed to Messrs. Dunlop and Bell of Liverpool from 1918/24. He then served as a seagoing engineer with Messrs. Alfred Holt and Company for ten years, obtaining a First Class Board of Trade Steam Certificate with Motor Endorsement. From 1933/36 he was chief engineer with the Ho-Hong Steam Ship Company of Singapore. He returned to the Blue Funnel Line for a further three years but in 1938 came ashore to join the Air Ministry and Ministry of Aircraft Production in London as engineer examiner to the chief examiner. In 1946 he was appointed examiner to the chief examiner at the Ministry of Aircraft Production in Liverpool, being associated with the Rootes Group, Armstrong Siddeley Motors Ltd., Rolls Royce Ltd., and D. Napier and Son Ltd. He was also charge inspector to the English Electric Co. Ltd. at Liverpool, where he did a great deal of work for the apprentices. He continued in these appointments until his death early in 1960, aged fifty-seven.

Mr. Joynson was elected to Membership of the Institute in 1944.

ERNEST ARTHUR SHEPHERD MACHON (Member 5092) died on 2nd March 1960, aged sixty-six years. He was educated at Bristol Grammar School and served his apprenticeship with J. Jefferies and Sons Ltd. of Avonmouth. On the outbreak of the first world war he joined the Royal Engineers, seeing service in Salonika and the Middle East, and returned to Bristol in 1919 with the rank of Captain.

He then joined Messrs. Charles Hill and Sons (Bristol City Line) as a marine engineer, rising to the rank of second engineer and obtaining an Extra First Class Board of Trade Certificate in 1924, when he joined the family firm of marine surveyors. Mr. Machon expanded the business to embrace marine engineering, being closely connected with the design and operation of sand dredgers in the Bristol Channel.

He held the appointment of British Corporation Register of Shipping and Aircraft surveyor and latterly American Bureau of Shipping surveyor. He was an Associate Member of the Institution of Mechanical Engineers and of the Royal Institution of Naval Architects; and had been a Member of the Institute of Marine Engineers since 1924.

ANDERSON MACPHEE (Member 1469) died on 19th January 1960, in his eightieth year. He had been connected with the Institute since 1900, having been elected a Graduate in that year, transferred to Associate five years later, and to full Membership in 1912. He was elected to Honorary Life Membership by the Council on 27th July 1959 in recognition of this long association.

He was apprenticed to G. and J. Weir Ltd., of Glasgow, and later to Siemens Brothers and Co. Ltd., Woolwich, from 1897/03, and then spent fourteen months at sea with the British India Steam Navigation Co. Ltd. He joined the Glasgow firm, Stewart Consulting Engineers, as a marine heating and ventilating engineer and then went to Philadelphia to become manager of the marine department of the Schute and Koerting Company. At the end of the first world war he left Philadelphia to be managing director of the Audale Engineering Company of Lansdale, Pennsylvania, a firm that manufactured marine equipment. In 1929 he was appointed the British agent for pulverized fuel and boiler fire equipment of the Kennedy Van Saun Manufacturing and Engineering Corporation, New York and London. He remained with this company until 1945 when he became consultant to the Mitchell Engineering Co. Ltd., London, his final appointment, which he resigned in 1954.

Mr. MacPhee had many of his own designs patented and some of these are still being manufactured by the Mitchell Engineering Company.

William Sellar (Member 4861) was born at Nigg, near Aberdeen, in 1885, and was educated at the Robert Gordon Technical College in Aberdeen, where he was awarded a Gold Medal for his work during the session 1901/02. He was apprenticed with Clyne, Mitchell and Co. Ltd., of the same town and then obtained drawing office experience with Hall, Russell and Co. Ltd., Aberdeen, David Rowan and Co. Ltd., Glasgow, and Ferguson Brothers Ltd., Port Glasgow.

In May 1915 he joined Harland and Wolff Ltd. at Belfast, in the engineering drawing office, and apart from a period of five years or so, from 1919/24, when he served in the marine department of Metropolitan Vickers Electrical Co. Ltd., as an engineer draughtsman, his connexion with the firm was unbroken until his sudden death on 15th January 1960.

In the service of Harland and Wolff Ltd. he was successively a senior draughtsman and a section leader—for a time assistant to the chief—in the Diesel engine drawing office; later he was employed in the engineering design office, engaged chiefly on development work.

Mr. Sellar joined the Institute as a Member in 1923, the year in which he presented a paper entitled "A Basis for the Explanation of Marine Gear Troubles", for which he was awarded the Denny Gold Medal.

He was an outstanding member of the Belfast North Branch of the Association of Engineering and Shipbuilding Draughtsmen and he frequently gave lectures at their meetings and to other engineering societies.

His love of nature, combined with keen powers of observation, caused him to take up art, and he was for many years a member of the Royal Ulster Academy of Art, exhibiting work in pastel, oil and water colour.

Einar Peter Christian Stahl (Member 15091), director for Marine Engineering Instruction, Mercantile Marine, Copenhagen, died on 29th January 1959, aged sixty-one.

He was an engineer cadet at the Royal Naval College, Copenhagen, from 1917/22, and served for the next six years as engineer officer in ships of the Royal Danish Navy. He then took a three-year course for engineer constructors at the Royal Technical College and from 1932/43 served as engineer constructor, first in coastal defence and later on materials inspection at the Royal Naval Dockyard. In May 1943 he became director in charge of the examination and education of engineers in the Danish Mercantile Marine.

Mr. Stahl joined the Institute as a Member in 1954; he was also a Member of the Institution of Danish Civil Engineers.

Ilmar Turn (Associate Member 18659) was born in Estonia in 1911 and attended the technical high school in Tartu. He was apprenticed at the Tallinn Harbour Works and then served as assistant engineer in Estonian ships for three years. He was more or less continuously at sea from 1941 until his death, obtaining a Second Class Australian Certificate in Melbourne in 1952 and a First Class Ministry of Transport Certificate in London in 1956. He joined the Institute in 1957.

Mr. Turn then sailed as chief engineer for Coulanthros Ltd., Sir R. Ropner and Co. Ltd., and finally in the m.v. *La Cordillera* owned by Buries Markes Ltd.; he died in this ship on 6th March 1960 and was buried in Sapele, Nigeria.